VENTILATION HANDBOOK

LIVESTOCK AND POULTRY

Engineering Branch

Province of British Columbia
Ministry of Agriculture and Food

VENTILATION HANDBOOK

LIVESTOCK AND POULTRY

by

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NOMENCLATURE

Units

SI

mm	=	Millimeter
m	=	Meter
g	=	Gram
kg	=	Kilogram
L	=	Litre
s	=	Second
h	=	Hour
J	=	Joule
W	=	Watt (= J/s)
N	=	Newton
Pa	=	Pascal
°C	=	Degrees Celsius
°K	=	Degrees Kelvin

Imperial

in	=	Inch
ft	=	Foot
lb	=	Pound
gal	=	Gallon (U.S.)
sec	=	Second
min	=	Minute
hr	=	Hour
wk	=	Week
Yr	=	Year
hp	=	Horse Power
Btu	=	British Thermal Unit
RPM	=	Revolutions per minute
cfm	=	Cubic feet per minute
gpm	=	Gallons (U.S.) per minute
ppm	=	Parts per million
psi	=	Pounds per square inch
°F	=	Degrees Fahrenheit
°R	=	Degrees Rankine

Symbols

A	=	Area
BSHL	=	Building sensible heat load
BTHL	=	Building total heat load
C_d	=	Coefficient of discharge
c_p	=	Specific heat
CH_4	=	Methane
CO	=	Carbon monoxide
CO_2	=	Carbon dioxide
d	=	Density
da	=	Dry air
dt/dx	=	Temperature gradient
E	=	Efficiency
E_c	=	Evaporative cooler efficiency
ECP	=	Evaporative cooler performance
h	=	Enthalpy
h_a	=	Enthalpy of dry air
h_g	=	Enthalpy of water vapour
H_20	=	Water
Hr	=	Heat recovery ratio
I	=	Inlet opening
k	=	Thermal conductivity
K	=	Coefficient
L	=	Length
LHL	=	Latent heat load
m	=	mass
M	=	Molecular weight
ML	=	Building moisture load
MP	=	Rate of moisture production
NH_3	=	Ammonia
P	=	Pressure

Symbols

P_w	=	Water vapour pressure
P_s	=	Saturated vapour pressure
POW	=	Power
q	=	Heat conduction rate
q_a	=	Animal heat production
q_b	=	Building heat loss
q_m	=	Mechanical heat production
q_s	=	Sensible heat production
q_{sh}	=	Supplemental heat requirement
q_v	=	Ventilation heat loss
Q	=	Airflow rate
R	=	Gas constant
R	=	Resistance value of insulation (Imperial)
RH	=	Relative humidity
RSI	=	Resistance value of insulation (SI)
SHL	=	Sensible heat loss
t	=	Temperature (conventional scale)
t_C	=	Celcius temperature
t_F	=	Fahrenheit temperature
T	=	Temperature (absolute scale)
T_K	=	Kelvin temperature
T_R	=	Rankine temperature
THL	=	Total heat loss
v	=	Volume
v_s	=	Specific volume
V	=	Velocity
W	=	Humidity ratio
Σ	=	Net sum

Note on Subscripts

Considerable use of subscript notation is made throughout the text of this handbook. Subscripts are attached to mathematical symbols (abbreviations) in order to identify a special case which may be associated with the symbol. Wherever possible, the first letter of the special case is used. For example:

- t (temperature) is subscripted "t_C" to indicate temperature in degrees Celsius.

- h (enthalpy) is subscripted with an "i" to indicate the enthalpy of air inside the building, h_i.

There are four commonly used subscripts which refer to the state or the location of the variable which is subscripted.

1 = at state one.
2 = at state two.
i = of the inside air.
o = of the outside air.

CHAPTER 1 INTRODUCTION

Advances in the fields of animal science and nutrition have developed to the stage where further improvements in animal production are likely to be slight. Control of the animal livestock environment is the next threshold to increased productivity of livestock and poultry.

Yet environmental control in confined animal environments is often very poor. Improvements in this area will certainly lead to increases in animal production. The full advantage of high quality feeds and special breeding can be capitalized only by maintaining optimum environmental conditions for the animal.

The producer who obtains the best feed conversion ratio will always have the competitive edge. Good heating and ventilation design for confined livestock production is essential if low feed conversion ratios are desired. Controlled environment means planning, design, and installation of a heating, ventilation and air-conditioning system which meets the physiological needs of the animal or bird.

This handbook is written for persons engaged in the practice of heating and ventilation design for animal rearing facilities. The ventilation designer must have a firm understanding of the scientific principles which are important to his discipline. No single publication could hope to address all of the scientific and practical matters which are related to heating and ventilation and this handbook is not a totally comprehensive manual of heating and ventilation. Rather, it is a handy reference guide to the principles and practice of heating and ventilation for animal housing.

Chapters 2 and 3 present the essential fundamentals of psychrometrics and animal physiology. Additional understanding and reference information in these areas should be obtained by the designer. Every attempt is made to provide useful information and design data in these sections. Tables of heat and moisture production, along with recommended temperatures, represent a compilation of the best data available from various sources.

Additional chapters deal with specific areas of ventilation system design: fans, inlets, cooling, heating systems, etc. are discussed separately. The purpose of separating ventilation design into

discrete subject areas is to aid the reader in finding useful information as quickly as possible. Don't design a ventilation system as separate parts which will hopefully work well together - design it as a single system with properly matched components.

A chapter on Trouble Shooting is provided to aid in the evaluation of existing livestock facilities. The trouble shooting hints are valuable as a list of possible source problems.

The handbook concludes with a chapter containing design examples which put into practice the theories and suggestions contained in the earlier chapters.

A bibliography is presented to aid the designer in finding useful reference information.

Note on Use of Units

It is strongly recommended that the designer use caution when solving ventilation problems with the use of either Imperial or SI (metric - International System) units. Confusion and error inevitably arise when conversion tables are used to switch between systems midway through a problem. The best practice is to use one system throughout.

Both systems are based on different premises (base-lines, datum, etc.) and direct conversion of working data sometimes leads to error. It is preferable that the designer become comfortable with one system before attempting to be an expert with both. Furthermore, solutions are much easier to follow when mixed units are not used.

Either system is equally functional for problem solving and selection of units is largely a matter of industry standard and personal preference. In Canada the shift to metric is largely complete and the SI system is the new industry standard; Imperial units (still used widely in the United States) are subordinated in this handbook by parentheses. All illustrations are presented in metric units.

TABLE OF APPROXIMATE CONVERSIONS

MULTIPLY	BY	TO OBTAIN
Length		
ft	0.3048	m
in	25.4	mm
Area		
ft^2	0.0929	m^2
in^2	0.645×10^{-3}	m^2
Volume		
ft^3	7.481	gal
ft^3	28.32	L
ft^3	0.02832	m^3
gal (U.S.)	3.785	L
Energy, Heat, Power		
Btu	1055	J
Btu	2.931×10^{-4}	kW-h
Btu/lb	2326	J/kg
Btu/lb-°F	4187	J/kg-°C
Btu/hr	0.2931	W
hp	2546	Btu/hr
hp	745.7	W
R(°F-hr-ft^2/Btu)	0.1763	RSI(°C-m^2/W)
W	1	J/s
J	1	kg-m^2/s^2

MULTIPLY	BY	TO OBTAIN
Force		
lb	4.448	N
N	1	kg-m/s^2
Mass		
lb	453.5	g
lb	6999	grains
Flow Rate		
cfm	0.472	L/s
gpm (U.S.)	0.0631	L/s
Velocity		
ft/min	5.08×10^{-3}	m/s
ft/sec	0.3048	m/s
mph	1.61	km/h
Pressure		
inches water column	0.0361	psi
inches water column @ 15.5°C	248.8	Pa
psi	6895	Pa
N/m^2	1	Pa

CHAPTER 2 PRINCIPLES OF VENTILATION

DEFINITIONS

Heat is a form of energy that can be transferred from one body to another because of a temperature difference between the two bodies. Heat always flows from a warmer to a cooler body.

Temperature is a measure of a body's ability to transfer or receive heat. The temperature of a substance may be expressed in either relative or absolute units. The two most familiar measures of temperature are the Celsius and Fahrenheit scales.

The Celsius scale is based on the arbitrary assignment of 0 degrees to melting ice and 100 degrees to boiling water at one atmosphere pressure. The Fahrenheit scale sets the melting point of ice at 32 degrees and the boiling point at 212 degrees. Either scale may be converted to the other by means of the following expressions:

$$t_F = 1.8\, t_C + 32 \qquad \text{(Eq. 2.1)}$$
$$t_C = (t_F - 32)/1.8$$

t_F = Fahrenheit temperature, °F
t_C = Celsius temperature, °C

The second Law of thermodynamics leads to the conclusion that there is a "lowest conceivable temperature", known as **absolute zero**. The Kelvin and Rankine scales are used for expressing temperature in absolute units. Absolute zero occurs at -273.16 degrees on the Celcius scale and -459.69 degrees on the Fahrenheit scale.

$$T_K = t_C + 273.16 \qquad \text{(Eq. 2.2)}$$

$T_R = t_F + 459.69$
T_K = Kelvin temperature, °K
T_R = Rankine temperature, °R

Specific heat capacity is the quantity of heat required to raise a unit weight of a material one degree. Units of specific heat capacity are kJ/kg-°C or Btu/lb-°F.

Sensible heat is a measure of the heat energy which is associated with a change in the temperature of a material.

$$q_s = mc_p(t_1 - t_2) \qquad \text{(Eq. 2.3)}$$

q_s = total sensible heat, kJ or Btu
m = mass of material, kg or lbs.

c_p = specific heat of the material, kJ/kg-°C or Btu/lb-°F

t_1 = temperature of material at condition 1, °C or °F

t_2 = temperature of material at condition 2, °C or °F

1 Btu = 1.055 kJ.

Note: The symbol "q" in this publication refers to heat energy in absolute terms (kJ or Btu) **or** heat energy change per unit time (kW or Btu/hr). Most design problems require heat energy change per unit time.

Latent heat is energy which is absorbed or released when a material changes phase without a change in temperature. The three phases are solid, liquid, and gas.

Latent heat of fusion refers to the energy required for the solid to liquid phase change (e.g., from ice to water).

Latent heat of vapourization refers to the energy required for the liquid to vapour phase change. The latent heat of vapourization of water varies with temperature, however for design of animal ventilation systems it is usually taken as 2428 kJ/kg or 1044 Btu/lb of water.

When water vapour cools it returns to the liquid phase, or **condenses**. The **heat of condensation** is equal, but of opposite magnitude to the heat of vapourization.

The difference between sensible and latent heat is important and must be clearly understood by the ventilation designer; sensible heat is the energy required to raise temperature, whereas latent heat is the energy required to change the water from one phase to another. Livestock produce both kinds of heat. Latent heat is released into the air in the form of respired water vapour and sensible heat is transferred by the animal to the environment by conduction, radiation and convection.

Heat conduction is the exchange of heat between contacting bodies that are at different temperatures. The flow of heat is always from the warmer to the cooler body and follows a temperature gradient. The rate of heat conduction through a material is given by

$$q = -Ak(dt/dx) \qquad \text{(Eq. 2.4)}$$

q = heat conduction rate, W or Btu/hr

A = cross-sectional area normal to direction of heat flow, m^2 or ft^2

k = thermal conductivity of material W/m-°C or Btu/(hr-ft^2-°F/in).

dt/dx = temperature gradient °C/m or °F/in.

For positive heat flow (heat gain) the temperature gradient is negative. In most practical applications, thermal resistance or R-value, is used instead of thermal conductivity, k.

$R = L/k$ (Eq. 2.5)

R = thermal resistance per unit thickness of material, °C-m^2/W (RSI value) or °F-hr-ft^2/Btu (R value).

And heat conduction is given by

$$q = \frac{A_i}{R_i}(t_i - t_o) \quad \text{(Eq. 2.6)}$$

t_i-t_o = temperature difference between inside and outside, °C or °F

A_i = Area under consideration (wall, window, etc.), m^2 or ft^2

R_i = R value under consideration (wall, window, etc.).

L = thickness, m or inches.

The thermal conductivity of a material is experimentally obtained and varies with density, temperature, moisture content, size of fibres and the arrangement and degree of bond between fibres or particles. In practise, a single value for R or k is chosen based upon normal conditions and construction methods.

Convection is the transfer of heat by the mass movement of a fluid. In ventilation design the fluid is usually a gas, namely, air. The laws governing convective heat transfer are too complex to be described in detail here, however they are important.

Convective heat transfer is classified as either **natural** or **forced convection.** In natural convection, a hot surface will cause a temperature increase in the fluid in contact with it. The heated fluid will expand and rise due to its decreased density. The motion of the fluid along the surface carries away the heat energy imparted to it, hence the term "mass movement" or "heat and mass transfer".

The action of convection is resisted by the viscosity of the fluid. Shape and orientation of the surface are also important, as well as other fluid properties.

Forced convection is where fluid is pumped past the surface, such as the case when air flows through a duct or when a fan blows air against a wall. The potential for increased heat transfer in the case of forced convection is great.

It is recommended that the ventilation designer consult a textbook on the subject of convection for a greater understanding of the complex processes involved in this type of heat transfer. The American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) **Handbook of Fundamentals**, 1981 presents extensive equations for heat transfer by convection.

Thermal radiation is the exchange of thermal energy by electromagnetic waves

and follows the same laws as light: it is transmitted through space, it follows a straight-line path, and it may be reflected, transmitted or absorbed. When radiation is absorbed, it is converted to heat.

The rate of radiant emission varies with the temperature of an object and the characteristics of the body. A body radiates energy in all directions and energy is transferred between two bodies in both directions - not just from a warmer to a cooler body. The net transfer of energy is from warm to cool bodies and may be calculated using formulas found in any elementary heat transfer textbook.

Relative Humidity (RH) is the ratio of partial water vapour pressure (P_w) to the vapour pressure of saturated air (P_s) at the same temperature, and is expressed as a percent.

$$RH = 100(P_w/P_s) \qquad \text{(Eq. 2.7)}$$

RH = Relative Humidity, %.

P_w = actual water vapour pressure, N/m^2 (Pa) or lb/ft^2.

P_s = saturated vapour pressure at same temperature, Pa or lb/ft^2.

1 Pascal (Pa) = 1 N/m^2 unit of pressure.

PSYCHROMETRICS

Psychrometrics is the study of the physical and thermodynamic properties of moist air. An understanding of psychrometrics is essential to the design of any ventilation system, whether it be for man, animal or plant. Animal (and human) comfort is greatly influenced by the temperature and relative humidity of the surrounding air.

Ideal Gas Law

The thermodynamic state of moist air is, to a good approximation, defined by the ideal gas law.

$$Pv = mRT \qquad \text{(Eq. 2.8)}$$

P = absolute pressure, Pa or lb/ft^2

v = volume, m^3 or ft^3

m = mass, kg or lb

R = a gas constant J/kg-°K or ft-lb/lb-°R

T = absolute temperature, °K or °R.

R_a = gas constant dry air
= 287 J/kg-°K = 53.35 ft-lb/lb-°R

R_w = gas constant water vapour.
= 461 J/kg-°K = 85.78 ft-lb/lb-°R

Equation 2.8 holds true for a mixture of dry air and water vapour with negligible error at the temperature and pressure ranges typical for environmental control of agricultural structures. The gas constants R_a and R_w shown above are functionally accurate for purposes of ventilation design.

Humidity Ratio

Dalton's Law (Equation 2.9) states that the pressure exerted by each gas in a mixture of gases is independent of the presence of other gases and that tne sum of partial pressures in a mixture of gases is equal to the total pressure. For a mixture of air, a, and water vapour, w, we have

(Eq. 2.9)

$$P = P_a + P_w = \frac{m_a R_a T_a}{V_a} + \frac{m_a R_a T_a}{V_w}$$

V_a = volume of dry air, m^3 or ft^3

V_w = volume of water vapour, m^3 or ft^3

m_a = mass of dry air, kg or lb

m_w = mass of water vapour, kg or lb

P_w = pressure of water varpour, Pascals or lb/ft^2

P_a = pressure of dry air, Pascals or lb/ft^2

The **humidity ratio** is a useful parameter when one is concerned with weights of water vapour in the air. Livestock produce water vapour through respiration and values for this production are usually tabulated as kilograms or pounds water per day. Note that humidity ratio is independent of temperature.

(Eq. 2.10)

$$W = \frac{m_w}{m_a} = \frac{R_a P_w}{R_w P_a} = 0.622\left(\frac{P_w}{P_{at} - P_w}\right)$$

W = humidity ratio, kg/kg dry air or lb/lb dry air

P_{at} = atmospheric pressure, Pascals or lb/ft^2

Relative Humidity

Relative humidity (RH) has been previously defined (Equation 2.7) but should be recognized as being different from the humidity ratio. Since air, at different temperatures, holds different amounts of water, the relative humidity varies with temperature. Cold air brought in from the outside environment holds less water at 100% relative humidity than warm air in the barn.

Since animal comfort is related to relative humidity, it is important to control the animal environment to keep RH within a specified comfort range. High relative humidities will lead to condensation problems and poor evaporation of any pools of water which may be present in the animal environment. Low relative humidities, on the other hand, can lead to severe dust problems in the air and possible irritation of animal respiratory systems. An objective of any good ventilation system is to provide an adequate means of controlling relative humidity.

Specific Volume

From the ideal gas law the specific

volume (v_s) of dry air may be calculated

$$v_s = \frac{v}{m_a} = \frac{R_a T_a}{P_a} \qquad \text{(Eq. 2.11)}$$

In air conditioning work, the volume of air is given as cubic meters (or cubic feet) per kilogram (or pound) of dry air together with the mass of water vapour associated with it. The base of one kilogram or pound of dry air is used because the weight of air entering and leaving an air conditioning unit remain constant once steady flow conditions are established.

Temperature

Three different temperatures describe an air and water-vapour mixture; namely, dry-bulb (t_{db}), wet-bulb (t_{wb}), and dew-point (t_{dp}) temperatures.

Dry-bulb temperature is measured with an ordinary mercury-in-glass thermometer, thermocouple, or thermistor. These must be properly shielded from radiation and kept dry.

Wet-bulb temperature is measured with the same devices as dry-bulb temperature except that the bulb or junction is wrapped in a water-moistened wick and held in a moving stream of ambient air. A steady state is established where sensible heat from the stream of ambient air provides heat of vapourization. The wet bulb is cooled by the evaporation of water from the wick, thus lowering the wet-bulb temperature below the dry-bulb temperature. Relative Humidity is determined by observing the difference between wet and dry bulb temperatures.

A **sling psychrometer** is usually used to measure both dry and wet bulb temperatures simultaneously. Two mercury-in-glass thermometers are mounted in a hand-held sling. One thermometer is covered with a wetted wick. The sling psychrometer is spun in the air to provide an air stream over the wick for a period of 60 seconds or more. When the two thermometers record steady temperatures between successive readings, steady-state has been established and the wet and dry bulb temperatures are noted. The difference between the dry and wet bulb temperatures is often called the **wet-bulb depression.**

The Dew-point temperature is the temperature at which moisture starts to condense when air, at constant pressure and humidity ratio, is cooled. The dew-point temperature is defined as the temperature of saturated air which has the same vapour pressure as the moist air under consideration. The dew-point temperature is extremely important in livestock ventilation design. If surfaces in the animal environment happen to be at or below the dew-point temperature, then water

will begin to condense and pool. A common problem in animal livestock facilities is condensation on poorly insulated exterior walls and flooring.

Enthalpy

Enthalpy is the heat energy content of an air and water-vapour mixture. The energy contained in the moist air is both sensible heat (indicated by dry-bulb temperature) and latent heat of vapourization (energy content of the water vapour). The values of enthalpy are not absolute and are based, for convenience, on the difference in internal energy between a fixed condition or datum and the condition being considered. The datum for dry air is 0°C. In Imperial units the datum is 0°F. The datum for water is 0°C or 32°F. Units are kJ/kg dry air or Btu/lb dry air (dry air is abbreviated "da").

The enthalpy (h) is defined by the relation:

$$h = h_a + Wh_g \qquad \text{(Eq. 2.12)}$$

h = enthalpy of air-vapour mixture kJ/kg da or Btu/lb da

$h_a = c_p t_{db}$ = enthalpy of dry air

h_g = enthalpy of water vapour, kJ/kg water or Btu/lb water.

c_p = specific heat of dry air, approximately
= 1.007 kJ/kg-°C = 0.24 Btu/lb-°F

t_{db} = dry-bulb temperature, °C or °F

W = humidity ratio of air, kg/kg da or lb/lb da.

Approximate enthalpy relations for the range of temperatures between 0°C - 60°C or 32°F - 140°F are often used for the normal case of 1 atmosphere pressure:

(Eq. 2.13)

SI: $h = (1.007t_{db}-0.026) + W(2501+1.84t_{db})$

Imperial: $h = 0.24t_{db} + W(1061 + 0.444t_{db})$

The Psychrometric Chart

A psychrometric chart is a graphical representation of both physical and thermal properties of moist air. It provides a convenient graphical method of solving air-conditioning problems. It is a valuable tool for the ventilation designer since it greatly reduces the tedious mathematical formulation required for numerical solution of air-conditioning problems. Figures 2.1, 2.2, and 2.3 present psychrometric charts in both SI and Imperial units.

Coordinates on the psychrometric chart axes are dry-bulb temperature along the abscissa (horizontal axis) and humidity ratio on the ordinate (vertical axis). The other properties given are wet-bulb and dew-point temperatures, enthalpy, relative humidity and specific volume. The intersection of any two property

Carrier

PSYCHROMETRIC CHART

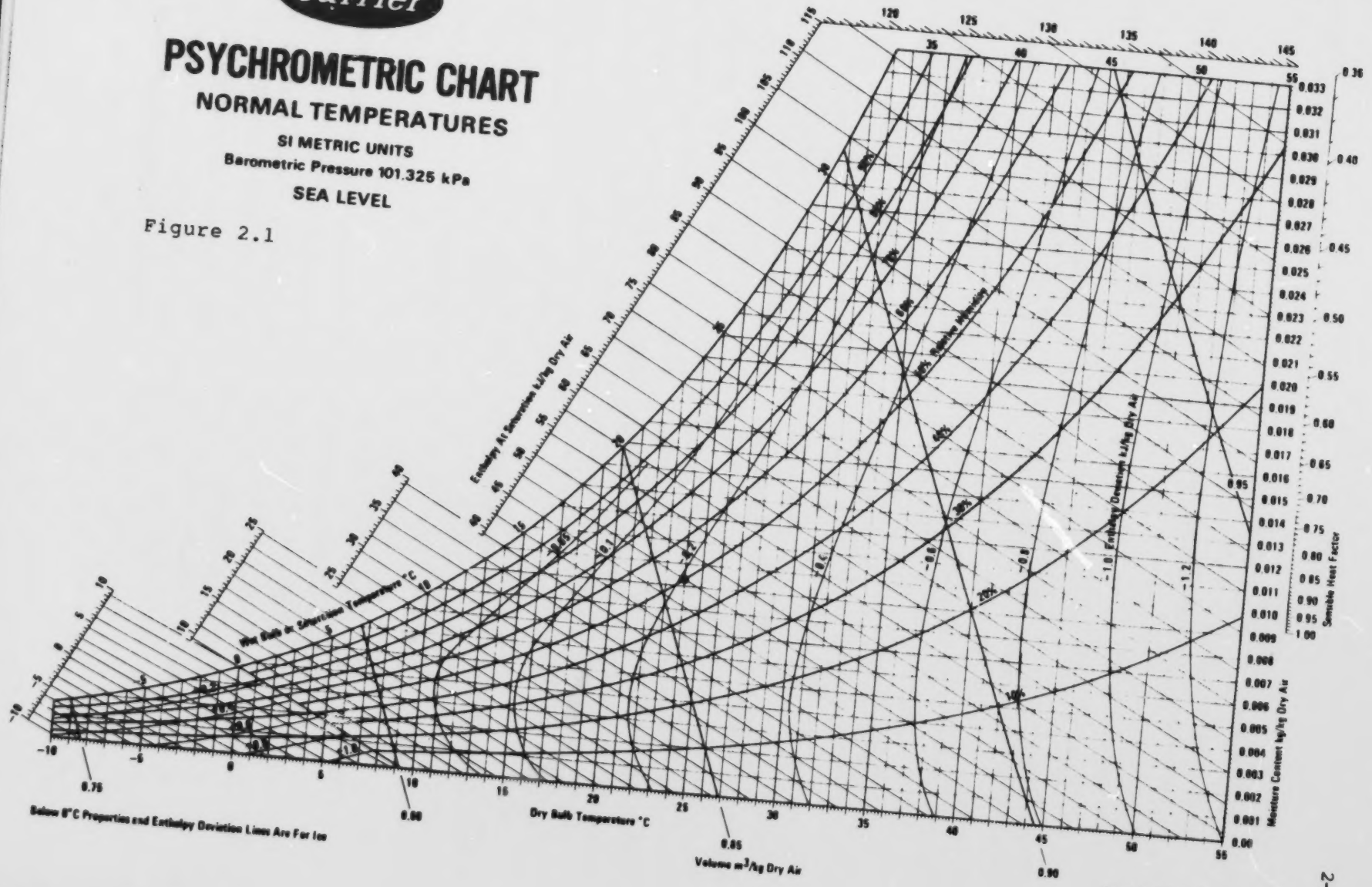
NORMAL TEMPERATURES

SI METRIC UNITS

Barometric Pressure 101.325 kPa

SEA LEVEL

Figure 2.1


Dry Bulb Temperature °C
Wet Bulb or Saturation Temperature °C
Enthalpy At Saturation kJ/kg Dry Air
Moisture Content kg/kg Dry Air
Sensible Heat Factor
Volume m³/kg Dry Air
Below 0°C Properties and Enthalpy Deviation Lines Are For Ice
-10 Enthalpy Deviation kJ/kg Dry Air
10% Relative Humidity

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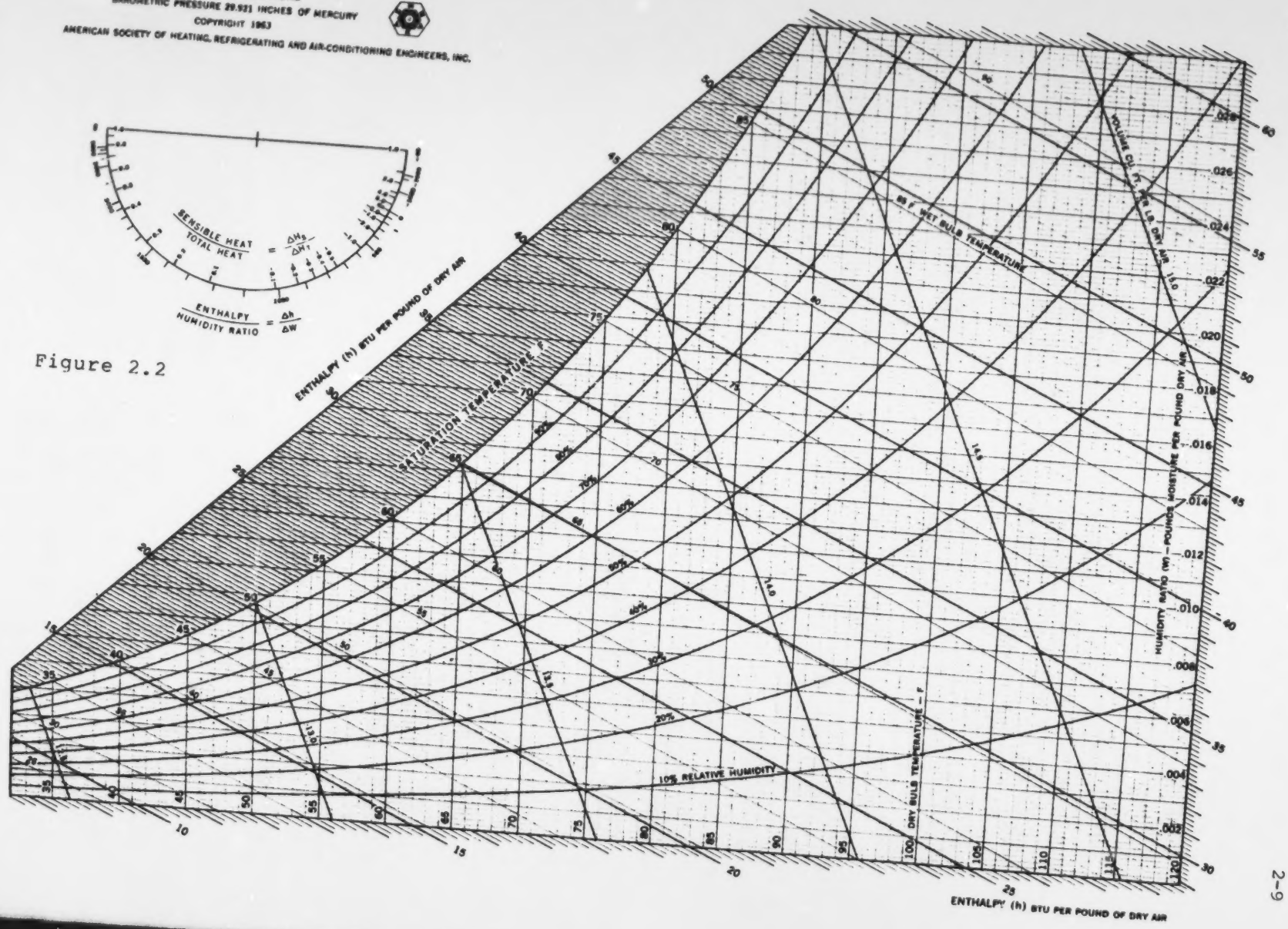
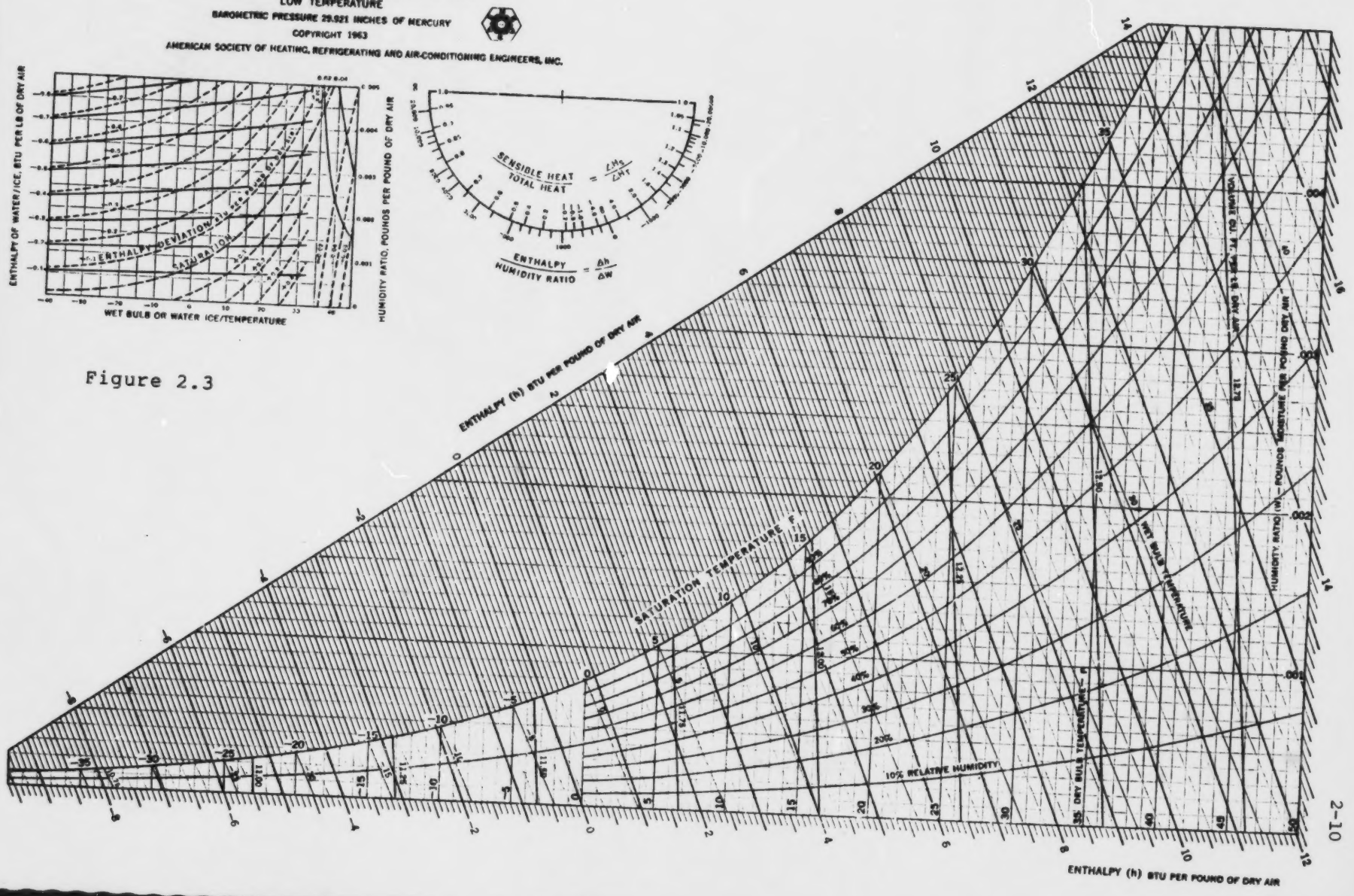

ASHRAE PSYCHROMETRIC CHART NO. 1
NORMAL TEMPERATURE
BAROMETRIC PRESSURE 29.921 INCHES OF MERCURY
COPYRIGHT 1953
AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, INC.
SENSIBLE HEAT / TOTAL HEAT = ΔHs / ΔHt
ENTHALPY / HUMIDITY RATIO = Δh / ΔW
ENTHALPY (h) BTU PER POUND OF DRY AIR
SATURATION TEMPERATURE - F
DRY BULB TEMPERATURE - F
HUMIDITY RATIO (W) - POUNDS MOISTURE PER POUND DRY AIR
VOLUME CU. FT. PER LB. DRY AIR
85 F WET BULB TEMPERATURE
10% RELATIVE HUMIDITY

Figure 2.2


ASHRAE PSYCHROMETRIC CHART NO. 2
LOW TEMPERATURE
BAROMETRIC PRESSURE 29.921 INCHES OF MERCURY
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AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, INC.
ENTHALPY OF WATER/ICE, BTU PER LB OF DRY AIR
ENTHALPY DEVIATION
SATURATION
WET BULB OR WATER ICE/TEMPERATURE
HUMIDITY RATIO, POUNDS PER POUND OF DRY AIR
SENSIBLE HEAT / TOTAL HEAT = Δhs/Δht
ENTHALPY / HUMIDITY RATIO = Δh/Δw
ENTHALPY (h) BTU PER POUND OF DRY AIR
SATURATION TEMPERATURE F
10% RELATIVE HUMIDITY
WET BULB TEMPERATURE
DRY BULB TEMPERATURE F
VOLUME CU. FT. PER LB. DRY AIR
HUMIDITY RATIO (w) POUNDS MOISTURE PER POUND DRY AIR
ENTHALPY (h) BTU PER POUND OF DRY AIR

Figure 2.3

lines fixes a given state, and all other properties can be read. Figure 2.4 shows the properties which can be obtained from the psychrometric chart.

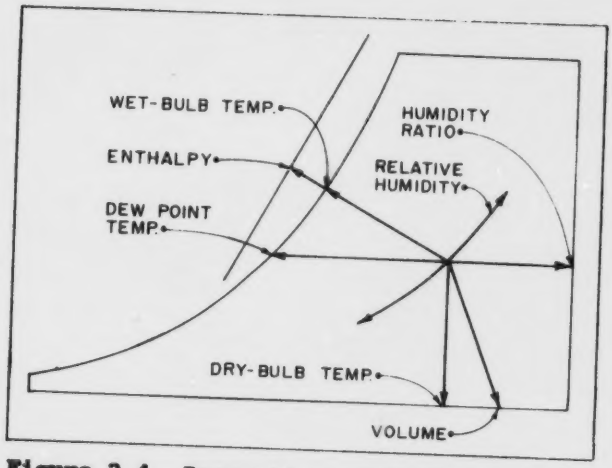


Figure 2.4 Properties of moist air on a psychrometric chart.

Air-Water Vapour Mixture Processes

The design of livestock housing usually involves some modification of environmental conditions for optimum production. Air conditioning processes such as heating, cooling, humidifying or dehumidifying and mixing of various air volumes may be required.

Sensible heating is the addition of heat to air without a change of humidity ratio. On the psychrometric chart the process takes place along a horizontal line and moves left to right. Sensible heating may be calculated from the mass and the enthalpy change of the air.

$$ {}_1q_2 = m(h_2 - h_1) \qquad \text{(Eq. 2.14)} $$

${}_1q_2$ = heat added in going from state 1 to state 2, kJ or Btu

m = mass of dry air, kg or lb

$h_2 - h_1$ = enthalpy difference of air in going from state 1 to state 2, kJ/kg da or Btu/lb da.

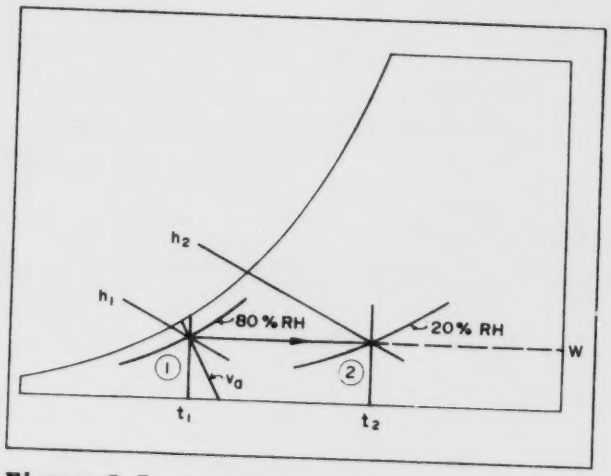


Figure 2.5 Sensible heating process.

Where a continuous process is being considered, express sensible heat loss or gain per unit of time:

$$ q = \frac{Q(h_2 - h_1)}{v_s} \qquad \text{(Eq.2.15)} $$

q = heat added, kW or Btu/min.

Q = air flow volume, m^3/s or ft^3/min.

v_s = specific volume, m^3/kg da or ft^3/lb da.

Sensible cooling is cooling of air at a constant humidity ratio and is represented on a psychrometric chart by moving on a horizontal line from right to left. Sensible cooling is often accomplished by means of passing an air stream over a cooling coil having a surface temperature above the dew-point temperature of the air (otherwise condensation and removal of latent heat will occur). Equations 2.14 and 2.15 are applicable to either sensible heating or cooling; the sign of the result indicates heat gain or loss.

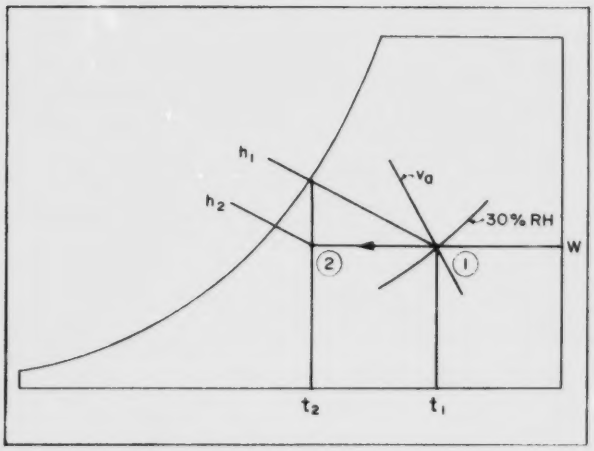


Figure 2.6 Sensible cooling process.

Evaporative cooling is an adiabatic (no heat loss or gain) saturation process (process of increasing humidity ratio) and follows upward along a constant wet-bulb temperature line. The sensible heat contained in an air stream which passes over a wetted surface converts water to vapour. This causes a decrease in the dry-bulb temperature of the air. Evaporative cooling is well suited to hot, dry climates where a large wet-bulb depression provides for a large temperature drop without raising relative humidity to unsatisfactory levels. For additional details see chapter 10.

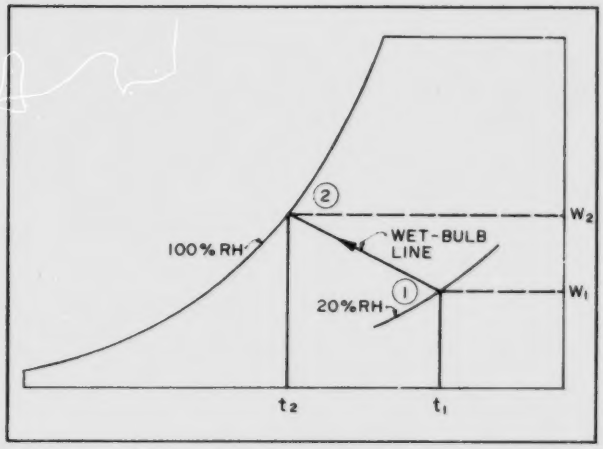


Figure 2.7 Evaporative cooling process.

Heating and Humidifying

As animals respire they produce heat and water vapour. Both sensible and latent heat are added to ventilating air as it moves through the livestock facility. The process is known as heating and humidifying.

Inspection of figure 2.8 shows that two separate calculations can be made: calculation of the sensible heat component, and calculation of the latent heat component.

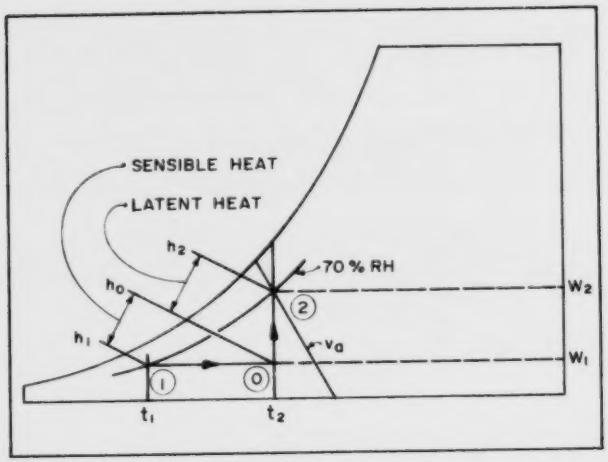


Figure 2.8 Heating and humidifying process.

1. Sensible heat added by livestock and supplemental heating system (if any): proceeds horizontally along the constant humidity ratio line.

   $_1q_0 = m(h_0-h_1)$

2. Latent heat added by livestock: proceeds vertically along constant dry-bulb temperature line.

   $_0q_2 = m(h_2-h_0)$

INSULATION

Insulation is used in poultry and livestock facilities for four important reasons.

1. Insulation reduces conductive heat loss in winter and heat gain in summer. Energy inputs for heating or cooling are thereby reduced.

2. Insulation reduces condensation on walls and other surfaces and makes a drier barn.

3. Insulation provides an extra measure of fire resistance (fire walls, etc.).

4. Temperature extremes of interior surfaces are reduced, thereby reducing radiation heat loss or gain.

Heat Loss

Proper insulation in a farm building will reduce heat losses caused by:

- conduction through walls, ceilings and foundations;
- Radiation
- convection on walls (interior and exterior) and ceilings;
- Air leakage; tight construction, with a sealed vapour barrier near the interior wall surface will minimize air leakage losses.

Condensation

Condensation can be a serious problem in agricultural structures. High relative humidities (and often low inside temperatures in livestock facilities) can cause condensation on cool wall and ceiling

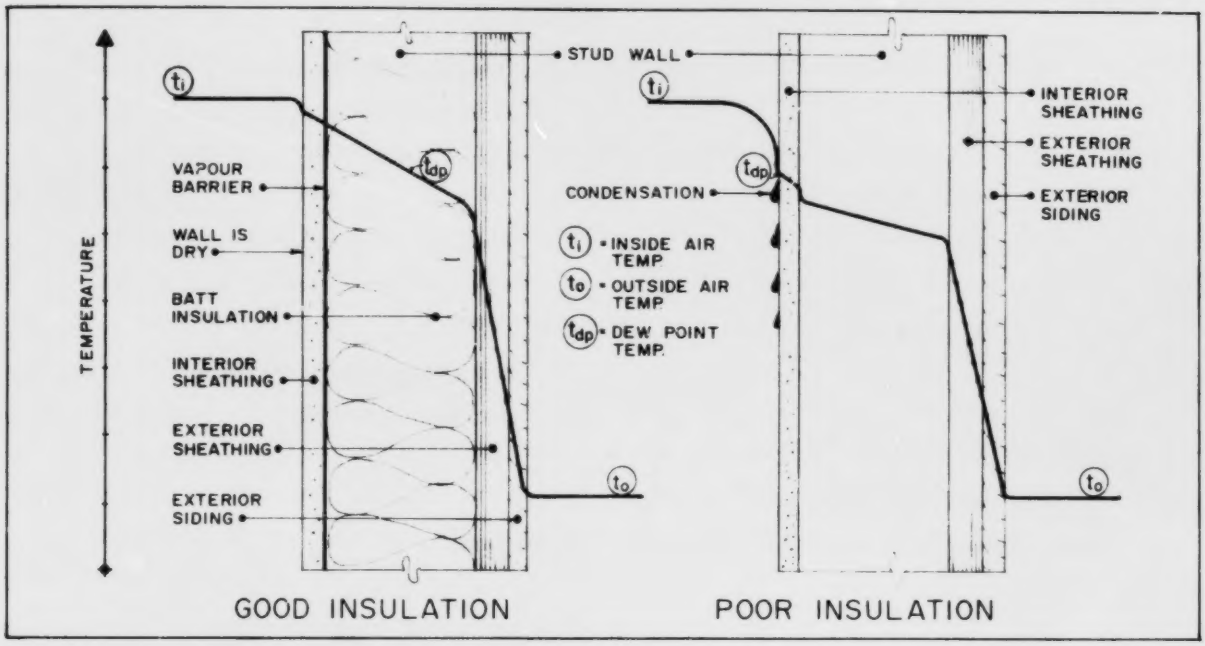


Figure 2.9 Temperature profile through a well and a poorly insulated wall. Condensation forms on a poorly insulated wall surface since the surface temperature is below the dew-point temperature. A vapour barrier prevents migration of moist air through the wall.

surfaces. If wall and ceiling surfaces are kept warm with good insulation, the air in contact with them remains above the dew-point and wetting is prevented. Wet barns are costly to heat since energy must be consumed to dry out the barn before temperature can be controlled.

Figure 2.9 shows how a poorly insulated wall has a wet interior surface, while a well-insulated wall will remain warm and dry. The well-insulated wall surface temperature is above the dew-point temperature and no condensation occurs.

It is also interesting to note the temperature gradient through the walls shown in figure 2.9. At some point within the wall, the temperature profile passes through the dew-point temperature. If no vapour barrier is present, then moist inside air will migrate through the wall surface and condense inside the wall. A properly sealed vapour barrier is essential to prevent moisture build-up in the wall's insulation. Wet insulation has greatly reduced insulative properties and wet wall materials are subject to rapid deterioration and rotting.

Heat Loss Calculation

The total insulation value and heat loss for a particular construction can be

estimated using relatively simple procedures. A number of good publications exist which describe methods of computing heat loss for a building. **Insulation in Farm Buildings**, Publication 1601/E, Agriculture Canada, 1983, is recommended as a good example.

Fibrous Blanket Insulation

Mineral wool and glass fibre: Both of these are commonly used in all types of construction and are relatively low in cost. They are also fire resistant and are effective insulators if properly installed. Materials are fibreglass, mineral wool or cellulose fibres, and are available in batts or blankets. Thicknesses range between 25 - 200 mm (1" and 8") and widths are designed to fit 400 mm, 600 mm, or 1200 mm (16", 24", or 48") stud centres. Some of the batts have a foil or paper backing which acts as a vapour barrier, however a proper polyethylene barrier is also required. "Friction-fit" insulation is an improved form of glass fibre blanket insulation and comes without a paper cover. This product is easier to work with and fits between studs and joists better than the old paper-backed materials.

Loose-Fill Insulation

Loose mineral wool can be purchased in bag form. It is more expensive than blankets but has the advantage of being able to be poured or blown into difficult to reach locations.

Vermiculite is expanded mica which can be poured in tight or confined spaces (such as light-aggregate concrete blocks) more easily than loose mineral wool.

Cellulose fibre is a macerated wood pulp or newsprint paper treated with fire retarding chemicals.

Pulverized polystyrene plastic foam in granular form can be used as a loose fill insulation. It has less tendency to absorb moisture than other loose fill materials.

Rigid Plastic Foam Insulation

Polystyrene foam is produced in "beadboard" or "extruded" insulation board form. Beadboard can be used in contact with dry soil, such as under light-duty concrete floors (farrowing creeps, for example) and for foundation perimeter insulation. When used for foundation perimeter insulation it should be protected with a high density asbestos cement board or pressure treated plywood for resistence to rodents and weather.

Extruded polystyrene board is a denser product with a smooth waterproof skin making it superior in wet locations. It

is frequently used to support and insulate under concrete floors which bear heavier loads such as tractors.

Polyurethane foam comes in sheets similar to polystyrene boards or can be foamed in place using special equipment. Some techniques of foaming in place involve spraying the polyurethane directly on the inside surface of the wall to be insulated. While polyurethane provides excellent insulative properties it **poses a fire hazard when exposed.** The farm insurance agency should be contacted before considering use of this material. The building's insurability against fire should be checked. Many foams must be covered with fire resistant materials such as plywood or gypsum board. A vapour barrier is recommended for use with polyurethane foam since the foamed in place product is not completely impermeable to moisture.

Recommended Insulation Levels

Regardless of the type used, a minimum level of insulation is required. Some construction methods will not yield adequate insulation and should be avoided. Block wall construction -even with loose fill insulation- is not adequate and should not be used for livestock or poultry facilities that require supplemental heating.

With energy costs continuing to escalate, and the nominal cost of increasing insulation value at the time of construction, the following insulation values are recommended:

- Perimeter foundation walls of heated buildings should be insulated with 50 mm (2") polystyrene beadboard or equivalent.

- Frame walls should be insulated to RSI 3.5 (R20) level for conditions in British Columbia. Colder climates may require more insulation.

- Ceilings should be insulated to RSI 4.9 (R28) level for conditions in British Columbia.

A tabulation of insulation materials and their respective RSI (R) values is contained in **Insulation in Farm Buildings**, Publication 1601/E, Agriculture Canada, 1983.

Vapour Barriers

Moist air tends to push through building materials from the warm side to the cold side and condenses at any location in the insulation where the temperature reaches dew-point. In livestock and poultry facilities, a vapour barrier on the inside of all building surfaces prevents moisture buildup in wall and ceiling

insulation. Such moisture buildup reduces insulating properties and causes wall rot.

A 0.1 mm (4 mil) clear polyethylene or fibre reinforced aluminium foil makes an effective vapour barrier. Large rolls should be used to minimize the number of joints.

A vapour-breathing material such as asphalt felt should be used under the exterior cladding of buildings. This is particularly true when the cladding is sheet metal. The asphalt felt will help prevent wind-driven rain and cold drafts from penetrating the exterior cladding while allowing water vapour in the wall to escape.

AIR DISTRIBUTION

Why Ventilate?

Ventilation is a continuous process which provides a healthy and comfortable environment for poultry, livestock and humans. The reasons for ventilation are summarized below:

- Provides fresh air and adequate oxygen for animals;
- Removes excess heat in hot weather;
- Removes odours and toxic gases created by animal manure;
- Removes air-born dust and disease organisms;
- Removes excess moisture from the air;
- Removes combustion products of unvented heaters;
- Creates ventilation patterns which induce desired dunging habits in animals such as swine.

Moisture Exchange

A primary purpose of winter ventilation is to remove excess moisture from the animal environment; that is, to control relative humidity.

As cold air is brought into the building it picks up water vapour and heat from animals, electrical equipment and supplemental heaters. The cold air holds little moisture (although RH may be above 80%). As the cold air warms, it can hold more moisture. The moisture holding capacity of air nearly doubles for every 11°C (20°F) increase in temperature. The ventilating air, brought in holding little water vapour, is exhausted from the building relatively moist.

Ventilation rates should be designed to maintain room air between 50% to 75% relative humidity. These figures strike a balance between excessive condensation

and excessive dust in the environment.

The requirements for moisture removal depend on animal type, size, stocking density, and manure management system. Evaporation from animal feces and urine can add substantially to the moisture control ventilation requirements.

Heat Exchange

In summer, a main purpose of ventilation is the removal of excess heat from the building. Total heat production from animals and equipment should equal total heat losses from the building if temperature is to be held constant. When heat production exceeds heat loss, ventilation should be increased.

Ventilating air is usually (except during a heat wave) cooler than inside air. Bringing in cooler outside air and venting hot inside air removes sensible heat. In cool weather, less ventilation is required to maintain heat balance.

HEAT AND MOISTURE BALANCE

The two primary purposes of ventilation are: (1) temperature regulation and (2) moisture (relative humidity) regulation. To regulate temperature, total heat losses must equal total heat gains; hence the term **heat balance**. To regulate moisture, animal moisture production must be matched by moisture removal; hence the term **moisture balance**.

Using heat energy and moisture balance concepts, two ventilation rates can be calculated that will maintain desirable room conditions. One of these ventilation rates is calculated to maintain indoor temperature at the desired level (heat balance). The other ventilation rate is calculated to maintain desired levels of relative humidity (moisture balance).

Ventilation for Heat Balance

(Eq. 2.16)

$$Q = \frac{v_s}{c_p(t_i - t_o)} \times (q_s - A/R(t_i - t_o))$$

Q = ventilating rate, m^3/s or ft^3/hr (divide by 60 for ft^3/min)

$t_i - t_o$ = indoor-outdoor temperature difference, °C or °F

c_p = specific heat of air, approximately 1000 J/kg-°C or 0.24 Btu/lb-°F

q_s = sensible heat production of animals, W or Btu/hr

A/R = building heat loss factor, W/°C or Btu/hr-°F

v_s = specific volume of air, m^3/kg or ft^3/lb.

Ventilation for Moisture Balance

(Eq. 2.17)

$$Q = v_s[MP/(W2-W1)]/3600$$

Q = ventilating rate, m^3/s or ft^3/sec (multiply by 60 for ft^3/min)

MP = rate of moisture production (animal respiration + evaporation) kgH_2O/h or lbH_2O/hr

W_1 = water vapour in incoming air, kgH_2O/kg da or lbH_2O/lb da

W_2 = water vapour in outgoing air, kgH_2O/kg da or lbH_2O/lb da.

Ventilation Curves

Using equations 2.16 and 2.17, a plot of ventilating rate versus outside temperature can be made. For ventilation design, the following parameters must be determined:

1. Minimum winter ventilating rate for moisture, contaminent gas, or odour control.

2. Maximum supplemental heating rate required for temperature control when ventilating at the minimum winter ventilation rate.

3. Maximum summer ventilation rate for temperature control. This occurs when the outside temperature rises to within 3-8°C (5-15°F) of the desired inside temperature. As outside temperature increases above this point, the ventilation required for temperature control increases dramatically and natural ventilation (or evaporative cooling - see Chapter 10) may need to be provided.

Figure 2.10 is an example of a ventilation curve for swine production. Ventilation curves are used to determine the "ideal ventilation rate curve." Using data found in tables 3.1 and 3.2 (Moisture Production, Sensible Heat Loss and Total Heat loss of Livestock or Poultry) the designer can use equations 2.15 and 2.16 to plot the curves of required ventilation for moisture and heat balance. A recommended minimum winter ventilation rate line is then added. The minimum rate is calculated from data found in table 4.1 (General Ventilation recommendations for Livestock and Poultry). From these three curves the ideal ventilation rate curve is determined (shown as a dash-dot line). The ideal ventilation rate is always the largest of the three curves at any point.

Careful consideration should be given to the results of the ventilation curve. It is entirely possible for the ventilation rate for moisture removal to be above the recommended minimum winter ventilation rate. In such a case the minimum winter ventilation rate should be selected as the higher of the two values.

The ideal ventilation rate curve is what a real ventilation system should approximate. The closer actual ventilation

is to the ideal ventilation, the better the design.

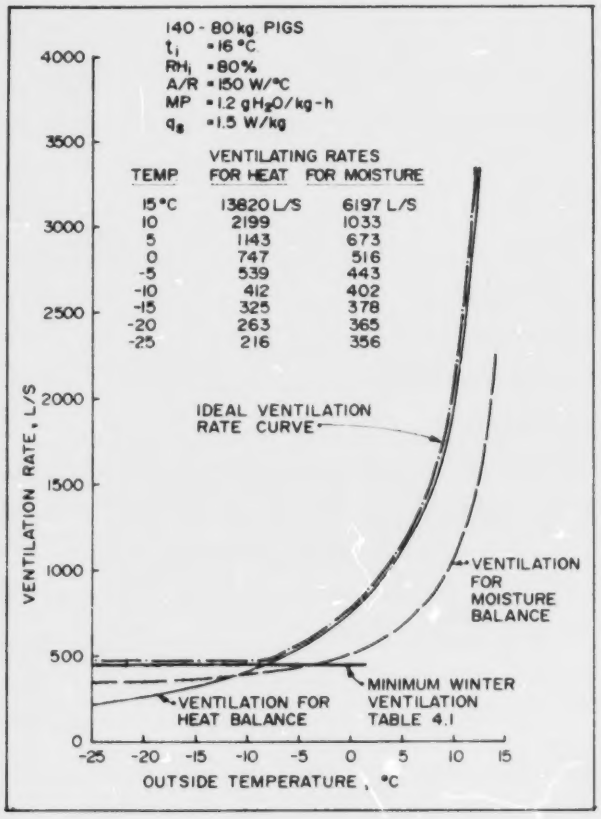


Figure 2.10 Ventilation curves for a 140 pig finisher barn.

Additional curves showing decreased stocking density and different inside and outside conditions provide the range of information which the designer needs to make intelligent decisions regarding equipment sizing and selection.

Specific methods for selection of supplemental heaters, fans, and other equipment are discussed in more detail under separate chapter headings.

COLD, MODIFIED AND WARM ENVIRONMENTS

Livestock confinement can be subdivided into three progressively more complex levels of control.

A **cold environment barn** may be described as an open barn such as a beef feedlot facility. Cold environment barns need little insulation and the inside temperature is little different from the outside temperature. Natural winds and convection currents provide the ventilation.

Quantities of ventilating air required are greater than in a warm barn since the ability of each unit volume of air to remove moisture is small when the temperature differential is low. The usual problems associated with a cold environment barn are keeping it dry (preventing condensation on walls and especially ceiling surfaces) and draft free, while still maintaining adequate air movement.

A **modified environment barn** is usually insulated but is not heated by mechanical means. A calf barn is a typical modified environment barn. It provides limited insulation in a semi-closed environment and may use mechanical ventilation to improve air exchange.

Because inside temperatures are higher than outside temperature, moisture removal is not as severe a problem as in

the case of a cold environment barn. Some insulation is required to prevent condensation due to cold wall and ceiling surfaces.

A **warm (or controlled) environment barn** provides full insulation, full mechanical ventilation, and supplemental heating (if required). The objective of a controlled environment barn is to create optimum conditions for animal growth. This handbook is primarily concerned with ventilation design for a controlled environment.

MECHANICAL VENTILATION

The arrangement of ventilation controls, fans, and inlets or outlets in a livestock confinement system has a significant effect. Different types of systems have different problems associated with them.

Positive pressure systems use fans to force fresh air into the building, creating a positive indoor pressure that pushes air out of the outlet openings. The air is distributed either by a baffle placed in front of the fan or by a duct having outlets evenly spaced along its length.

In the case of the duct, insulation is required to prevent condensation on the duct surface. Positive pressure systems have a tendency to force moist air into the walls and attic, so a proper vapour barrier is essential. Frost is also a problem. Air leaks around doors and windows cause these openings to become "frosted shut" in cold weather. In general, **positive pressure systems are not recommended.**

Positive pressure systems are usually used in older barns where it is easier to install pressure fan-inlets than air inlets for exhaust ventilation. The use of positive pressure ventilation should be restricted to remodeled buildings; never design a new facility for positive pressure ventilation.

Neutral pressure systems use fans on both inlets and outlets; building pressure is neutral since the air is powered in to the building, and powered (rather than pushed) out.

System components include both intake fans and exhaust fans. Examples of neutral pressure systems are Martin-Air, Fristimat, Aston and recirculating poly-tube systems.

Exhaust (or negative pressure) systems are the most common method of ventilation. Exhaust fans draw air out of the building, creating negative pressure in the building which causes fresh air to be sucked in through baffled air inlets. Provided that the fans and inlets are properly designed, exhaust systems distribute

fresh air uniformly and have good overall performance.

Ventilation by Displacement

This philosophy of ventilation is based on a "plug flow" concept. The idea is to bring air into the building at a low velocity through large vents on one wall. The low velocity air mass (or plug) then moves as a single non-turbulent unit across the structure and out the exhaust vents on the opposing wall. Unfortunately the theory only works for large volume ventilation flows (summer conditions). Ventilation systems designed on the displacement principle display extremely poor winter performance.

Dead spots and poor air distribution patterns are common in displacement ventilation systems. Mixing of the air with blending fans may be required to improve winter system performance (see Trouble Shooting, chapter 12).

Ventilation by Dilution (Mixing)

The idea behind the ventilation by dilution philosophy is that high velocity air inlets will cause mixing of incoming air with room air. The room odours, gases and water vapour will be diluted by the fresh air entering the structure.

The objective of ventilation by dilution is to cause complete mixing of fresh and room air without causing drafts. With proper inlet design, this objective can be achieved. Ventilation systems designed on the dilution philosophy are, in general, superior to displacement systems.

CHAPTER 3 ANIMAL PHYSIOLOGY

ANIMAL TEMPERATURE REGULATION

Livestock and poultry are known as "warm blooded animals". The proper term for warm blooded is **homeothermic**, which means simply that the animal is capable of maintaining a constant body temperature. The physiological mechanisms of homeothermy are important to the ventilation designer. By creating an environment for the animal which is well suited to its physiology, production gains will be maximized.

HEAT LOSS

Non-evaporative heat loss
(Sensible Heat Loss)

- Animal has no control over mechanisms of loss.
- Losses based on air and surface temperature in environment and animal skin temperature.
- Mechanisms of heat transfer are:
  - Radiation
  - Convection
  - Conduction
  - Excretion

Evaporative Heat Loss
(Latent Heat of Vapourization)

- Animal has some control over mechanisms of loss.
- Losses based on air temperature, relative humidity, and rate of perspiration and breathing.
- Mechanism of heat transfer is vapourization of water:
  - Cutaneous (perspiration through skin)
  - Pulmonary (rapid shallow breathing /panting)

Figure 3.1 Heat loss mechanisms for animal body temperature regulation.

Basal heat production is the amount of heat that an animal produces internally as a result of metabolic activity, i.e., the burning of fats, carbohydrates, and proteins in the food the animal eats. For homeothermy, the amount of basal heat production must be balanced by heat losses from the animal to the environment. Such heat losses can be broken down into two types: **evaporative heat loss** and **non-evaporative heat loss.** Figure 3.1 shows this breakdown.

The energy contained in the animal feed is converted to basal heat and productive work or weight gain. Through animal respiration, heat and moisture are added to the building environment.

When the animal defecates, the sensible heat contained in the urine and feces is transferred to the environment. Subsequent evaporation of animal feces and urine also adds moisture to the building environment.

A special case of heat transfer arises when air and environment temperatures are **greater** than the animal body temperature. Peak summer temperatures can result in sensible heat transfer from the environment to the animal. Under such conditions the animal must give up the excess heat by the mechanisms of evaporative heat loss alone.

Animal heat production can be estimated by the relation,

$$q_a = KM^{0.75}$$

q_a = animal heat production, Watts or Btu/hr

K = coefficient - 9 Watts or 16 Btu/hr

M = animal weight, kg or lb.

Heat and Moisture Production Tables

Use of Tables

Tables 3.1 and 3.2 present heat and moisture production data for various livestock and poultry. These data were compiled from various sources and represent the best information available at present. It is necessary to note that such data are not perfect; the numbers are only as good as the method employed to obtain them and are only applicable under the test conditions. The designer should regard the figures given in Tables 3.1 and 3.2 as good approximations and allow an extra margin for differences between experimental and commercial conditions.

The design problem may deal with a different classification of animal than is presented in the tables. In such cases it will be necessary to make a guess as to the classification which most closely approximates the animal or bird under consideration.

Specific moisture production (MP), specific sensible heat loss (SHL), and specific total heat loss (THL) data are given for livestock and poultry growing under various temperature conditions.

Animals not on a normal feeding level may have different heat and moisture losses. Animals on a maintenance diet, for example, may produce heat and moisture at as little as one-half of the rates stated. Maintenance feeding may be used in the case of sheep reared for fleece production, or during shipment of animals, or as a consequence of limited feeding.

The respective percentages of the building heat load which are sensible or latent will depend upon the methods of waste management in the facility. Feces and urine can be converted to water vapour by sensible heat. Such evaporation will reduce the sensible heat content of the air and increase the latent heat (moisture) content of the air. Total heat loss from the animal will remain the same. The evaporation of water can contribute to cooling during warm weather but during the winter such evaporation can lead to increased supplemental heating requirements.

Other factors can also be important. Huddling pigs lose less heat than piglets which are prevented from huddling or grouping. Conductive losses through floors can be substantially reduced by floor heaters.

Moisture production (MP) is given in units of grams of water produced per kilogram of animal liveweight per hour. The Imperial units are grains of water produced per pound of animal liveweight per hour. To calculate moisture production in kilograms or pounds use the following expressions:

(Eq. 3.1)

SI

ML = (no. of animals)(average wt. of animal) MP/1000

Imperial

ML = (no. of animals)(average wt. of animal) MP/6999

ML = building moisture load kgH_2O/h or lbH_2O/hr

MP = specific moisture production $gH_2O/Kg\text{-}h$ or grains $H_2O/lb\text{-}hr$

1 lb = 6999 grains

Sensible and total heat loads are given in units of watts per kilogram or in British thermal units per pound per hour. Total building heat load is found by a relation similar to Equation 3.1.

(Eq. 3.2)

BSHL = (no. of animals)(avg. wt. of animal)SHL

BTHL = (no. of animals)(avg. wt. of animal)THL

SHL = sensible heat load (tabulated)

THL = total heat load (tabulated)

BSHL = building sensible heat load, W or Btu/hr

BTHL = building total heat load, W or Btu/hr

Latent heat load can be found from moisture production.

(Eq. 3.3)

SI

LHL = 0.674 (MP)

Imperial

LHL = 0.149 (MP)

LHL = latent heat load, W/kg or Btu/lb-hr

TABLE 3.1

MOISTURE PRODUCTION (MP), SENSIBLE HEAT LOSS (SHL), AND TOTAL HEAT LOSS (THL) OF LIVESTOCK

Livestock	Temperature °C (°F)	MP $\frac{gH_2O}{kg-h}$ $\left(\frac{\text{grains } H_2O}{lb-hr}\right)$	SHL $\frac{W}{kg}$ $\left(\frac{Btu}{lb-hr}\right)$	THL $\frac{W}{kg}$ $\left(\frac{Btu}{lb-hr}\right)$	Ref.
Cattle					
Dairy cow					13
(stable)					
500kg (1100lb)	-1 (30)	0.77 (5.4)	1.9 (2.9)	2.4 (3.7)	
500kg (1100lb)	10 (50)	1.0 (7.0)	1.5 (2.3)	2.2 (3.4)	
500kg (1100lb)	15 (59)	1.3 (9.1)	1.2 (1.9)	2.1 (3.3)	
500kg (1100lb)	21 (70)	1.3 (9.1)	1.1 (1.7)	2.0 (3.1)	
500kg (1100lb)	27 (81)	1.8 (12.6)	0.6 (0.9)	1.9 (2.9)	
Beef cattle					13
(stable)					
500kg (1100 lb)	4 (39)	2.5 (17.5)	1.5 (2.3)	2.8 (4.3)	
Calves					
Ayrshire male					13
39kg (86lb) 8 days	3 (37)	0.7 (4.9)	2.5 (3.9)	2.9 (4.5)	
40kg (88lb) 14 days	3 (37)	0.7 (4.9)	2.4 (3.7)	2.8 (4.3)	
45kg (99lb) 25 days	3 (37)	0.7 (4.9)	2.6 (4.0)	3.0 (4.7)	
39kg (86lb) 8 days	23 (73)	0.7 (4.9)	2.0 (3.1)	2.4 (3.7)	
40kg (88lb) 14 days	23 (73)	0.7 (4.9)	1.9 (2.9)	2.3 (3.6)	
44kg (97lb) 24 days	23 (73)	0.7 (4.9)	2.0 (3.1)	2.4 (3.7)	
Fresian male					
100kg (220lb)	5 (41)	0.75 (5.2)	2.1 (3.3)	2.6 (4.0)	
100kg (220lb)	10 (50)	1.4 (9.8)	2.0 (3.1)	2.6 (4.0)	
100kg (220lb)	15 (59)	2.0 (14.0)	1.4 (2.2)	2.8 (1.3)	
100kg (220lb)	20 (68)	2.3 (16.1)	1.4 (2.2)	2.9 (4.5)	
Jersey					
(stable)					
16 wk	10 (50)	2.4 (16.8)	2.5 (3.9)	4.1 (6.4)	
32 wk	10 (50)	1.5 (10.5)	1.8 (2.8)	2.8 (4.3)	
48 wk	10 (50)	1.3 (9.1)	1.6 (2.5)	2.5 (3.9)	
16 wk	27 (80)	3.8 (26.6)	1.4 (2.2)	3.9 (6.1)	
32 wk	27 (80)	2.5 (17.5)	1.0 (1.6)	2.7 (4.2)	
48 wk	27 (80)	2.3 (16.1)	0.8 (1.2)	2.3 (3.6)	

Livestock	Temperature °C (°F)	MP $\frac{gH_2O}{kg\text{-}h}$ $\left(\frac{\text{grains } H_2O}{\text{lb-hr}}\right)$	SHL $\frac{W}{kg}$ $\left(\frac{Btu}{lb\text{-}hr}\right)$	THL $\frac{W}{kg}$ $\left(\frac{Btu}{lb\text{-}hr}\right)$	Ref.
Swine					
Sow and litter (solid floor)					13
177kg (390lb) 0 wk	16-27 (60-80)	1.8 (12.6)	1.3 (2.0)	2.6 (4.0)	
181kg (398lb) 2 wk	16-27 (60-80)	2.4 (16.8)	1.7 (2.6)	3.3 (5.1)	
186kg (410lb) 4 wk	16-27 (60-80)	2.6 (18.2)	1.7 (2.6)	3.5 (5.4)	
200kg (440lb) 6 wk	16-27 (60-80)	2.7 (18.9)	1.7 (2.6)	3.5 (5.4)	
227kg (500lb) 8 wk	16-27 (60-80)	2.6 (18.2)	2.1 (3.3)	3.9 (6.1)	
Nursery pigs					13
4-6kg (9-13lb)	29 (84)	1.7 (11.9)	2.2 (3.4)	3.3 (5.1)	
6-11kg (13-24lb)	24 (75)	2.2 (15.4)	3.1 (4.8)	4.5 (7.0)	
11-17kg (24-37lb)	18 (64)	2.2 (15.4)	3.5 (5.4)	5.0 (7.8)	
Growing - finishing pigs (solid floor)					13
20kg (44lb)	5 (41)	2.5 (17.5)	4.2 (6.5)	5.9 (9.2)	
	10 (50)	2.7 (18.9)	4.0 (6.2)	5.4 (8.4)	
	15 (59)	3.1 (21.7)	3.0 (4.7)	5.0 (7.8)	
	20 (68)	3.7 (25.9)	2.3 (3.6)	4.8 (7.5)	
	25 (77)	4.7 (32.9)	1.6 (2.5)	4.8 (7.5)	
	30 (86)	6.3 (44.1)	0.6 (0.9)	4.8 (7.5)	
40kg (88lb)	5 (41)	1.5 (10.5)	3.0 (4.7)	4.0 (6.2)	
	10 (50)	1.6 (11.2)	2.5 (3.9)	3.6 (5.6)	
	15 (59)	1.9 (13.3)	2.0 (3.1)	3.3 (5.1)	
	20 (68)	2.2 (15.4)	1.6 (2.5)	3.1 (4.8)	
	25 (77)	2.8 (19.6)	1.2 (1.9)	3.0 (4.7)	
	30 (86)	3.6 (25.2)	0.6 (0.9)	3.0 (4.7)	
60kg (132lb)	5 (41)	1.2 (8.4)	2.5 (3.9)	3.3 (5.1)	
	10 (50)	1.3 (9.1)	2.0 (3.1)	2.9 (4.5)	
	15 (59)	1.4 (9.8)	1.7 (2.6)	2.6 (4.0)	
	20 (68)	1.7 (11.9)	1.3 (2.0)	2.4 (3.7)	
	25 (77)	2.0 (14.0)	1.0 (1.6)	2.3 (3.6)	
	30 (86)	2.7 (18.9)	0.5 (0.8)	2.3 (3.6)	
80kg (176lb)	5 (41)	1.1 (7.7)	2.2 (3.4)	2.9 (4.5)	
	10 (50)	1.1 (7.7)	1.8 (2.8)	2.5 (3.9)	
	15 (59)	1.2 (8.4)	1.5 (2.3)	2.3 (3.6)	
	20 (68)	1.4 (9.8)	1.2 (1.9)	2.1 (3.3)	
	25 (77)	1.7 (11.9)	0.85 (1.3)	2.0 (3.1)	
	30 (86)	2.2 (15.4)	0.49 (0.8)	1.9 (2.9)	

Livestock	Temperature °C (°F)	MP $\frac{gH_2O}{kg-h}$ $\left(\frac{grains\ H_2O}{lb-hr}\right)$	SHL $\frac{W}{kg}$ $\left(\frac{Btu}{lb-hr}\right)$	THL $\frac{W}{kg}$ $\left(\frac{Btu}{lb-hr}\right)$	Ref.
100kg (220lb)	5 (41)	0.94 (6.6)	2.0 (3.1)	2.6 (4.0)	13
	10 (50)	1.0 (7.0)	1.6 (2.5)	2.3 (3.6)	
	15 (59)	1.1 (7.7)	1.3 (2.0)	2.0 (3.1)	
	20 (68)	1.2 (8.4)	1.1 (1.7)	1.9 (2.9)	
	25 (77)	1.4 (9.8)	0.80 (1.2)	1.8 (2.8)	
	30 (86)	1.8 (12.6)	0.49 (0.8)	1.7 (2.6)	
Gilts, sows and boars (solid floor)					13
140kg (308lb)	5 (41)	0.79 (5.5)	1.8 (2.8)	2.3 (3.6)	
	10 (50)	0.79 (5.5)	1.5 (2.3)	2.0 (3.1)	
	15 (59)	0.84 (5.9)	1.2 (1.9)	1.8 (2.8)	
	20 (68)	0.93 (6.5)	1.0 (1.6)	1.6 (2.5)	
	25 (77)	1.1 (7.7)	0.77 (1.2)	1.5 (2.3)	
	30 (86)	1.3 (9.1)	0.54 (0.8)	1.4 (2.2)	
180kg (396lb)	5 (41)	0.64 (4.5)	1.7 (2.6)	2.1 (3.3)	
	10 (50)	0.63 (4.4)	1.4 (2.2)	1.8 (2.8)	
	15 (59)	0.65 (4.5)	1.2 (1.9)	1.6 (2.5)	
	20 (68)	0.70 (4.9)	0.97 (1.5)	1.4 (2.2)	
	25 (77)	0.80 (5.6)	0.80 (1.2)	1.3 (2.0)	
	30 (86)	0.96 (6.7)	0.63 (1.0)	1.3 (2.0)	
Sheep					
Mature 60kg (132lb) (maintenance diet) Fleece length					13
shorn	8 (46)	0.33 (2.3)	2.4 (3.7)	2.6 (4.0)	
	20 (68)	0.40 (2.8)	1.5 (2.3)	1.7 (2.6)	
	32 (90)	0.79 (5.5)	0.81 (1.3)	1.3 (2.0)	
3cm (1.2in)	8 (46)	0.34 (2.4)	1.2 (1.9)	1.4 (2.2)	
	20 (68)	0.52 (3.6)	0.94 (1.5)	1.3 (2.0)	
	32 (90)	1.3 (9.1)	0.45 (0.7)	1.3 (2.0)	
6cm (2.4in)	8 (46)	0.40 (2.8)	1.0 (1.6)	1.3 (2.0)	
	20 (68)	0.76 (5.3)	0.68 (1.1)	1.2 (1.9)	
	32 (90)	1.4 (9.8)	0.29 (0.5)	1.2 (1.9)	

Livestock	Temperature °C (°F)	MP $\frac{gH_2O}{kg-h}$ $\left(\frac{\text{grains } H_2O}{lb-hr}\right)$	SHL $\frac{W}{kg}$ $\left(\frac{Btu}{lb-hr}\right)$	THL $\frac{W}{kg}$ $\left(\frac{Btu}{lb-hr}\right)$	Ref.
Sheep (cont'd)					
1.3 x maintenance diet					
Fleece length					
12cm (4.8in)	13 (55)	1.0 (7.0)	0.85 (1.3)	1.5 (2.3)	
	23 (73)	1.5 (10.5)	0.54 (0.8)	1.5 (2.3)	
Rabbits					
2.3kg (5lb)	10-16 (50-60)	4.3 (30)	3.9 (6.0)	5.7 (8.9)	8

TABLE 3.2

MOISTURE PRODUCTION (MP), SENSIBLE HEAT LOSS (SHL), AND TOTAL HEAT LOSS (THL) OF POULTRY

Poultry	Temperature °C (°F)	MP $\frac{gH_2O}{kg\text{-}h}$ $\left(\frac{\text{grains } H_2O}{\text{lb-hr}}\right)$	SHL $\frac{W}{kg}$ $\left(\frac{Btu}{lb\text{-}hr}\right)$	THL $\frac{W}{kg}$ $\left(\frac{Btu}{lb\text{-}hr}\right)$	Ref.
Laying hen					
White Leghorn	8 (46)	2.1 (14.7)	4.5 (7.0)	5.8 (9.0)	13
	12 (54)	2.8 (19.6)	4.0 (6.2)	5.9 (9.2)	
	18 (64)	2.9 (20.3)	3.9 (6.1)	5.8 (9.0)	
	28 (82)	3.8 (26.6)	3.2 (5.0)	5.8 (9.0)	
(including defecation & wasted drinking water)	-4 (25)	5.7 (39.9)	-	-	17
	2 (35)	6.2 (43.3)	-	-	
	7 (45)	5.9 (41.5)	-	-	
	16 (60)	6.6 (46.2)	-	-	
	27 (80)	7.9 (55.3)	-	-	
	35 (95)	8.4 (59.0)	-	-	
Broilers					
0.1kg (0.21b)	29 (84)	4 (28)	12 (19)	14 (22)	13
0.7kg (1.51b)	30 (86)	10 (70)	6 (9)	12 (19)	
	25 (77)	3 (21)	7 (11)	9 (14)	
	16 (61)	7 (49)	6 (9)	11 (17)	
1.1kg (2.41b)	30 (86)	10 (70)	4 (6)	11 (17)	
	19 (66)	1.8 (12.6)	7 (11)	8.9 (13.8)	
	16 (61)	4.5 (31.5)	3.6 (5.6)	7.2 (11.2)	
1.6kg(3.51b)	19 (66)	2.0 (14.0)	5.7 (8.8)	7.1 (11.0)	
2.0kg (4.41b)	19 (66)	1.4 (9.8)	4.8 (7.5)	5.8 (9.0)	
Pullets (reared for egg production)					
day old 0.036kg	21 (70)	5.6 (39.2)	9.7 (15.1)	16.0 (24.8)	20
3 wk 0.11kg	21 (70)	8.3 (58.1)	14.5 (22.5)	20.1 (31.2)	
8 wk 0.7kg	21 (70)	2.2 (15.4)	7.6 (11.8)	9.1 (14.1)	
12 wk 1.1kg	13 (55)	1.9 (13.3)	6.4 (9.9)	7.7 (12.0)	
16 wk 1.6kg	13 (55)	1.6 (11.2)	5.4 (8.4)	6.5 (10.1)	
18 wk 1.8kg	13 (55)	1.5 (10.5)	5.4 (8.4)	6.4 (9.9)	

Poultry	Temperature °C (°F)	MP $\frac{gH_2O}{kg-h}$ $\left(\frac{\text{grains } H_2O}{\text{lb-hr}}\right)$	SHL $\frac{W}{kg}$ $\left(\frac{\text{Btu}}{\text{lb-hr}}\right)$	THL $\frac{W}{kg}$ $\left(\frac{\text{Btu}}{\text{lb-hr}}\right)$	Ref.
Turkeys					
Toms					
0.1kg (0.21b)	35 (95)	15.6 (109.2)	6.1 (9.5)	16.8 (26.1)	13
0.2kg (0.41b)	32 (90)	9.8 (68.6)	6.3 (9.8)	12.9 (20.0)	
0.4kg (0.91b)	29 (84)	7.2 (50.4)	5.4 (8.4)	10.2 (15.8)	
0.6kg (1.31b)	27 (80)	4.1 (28.7)	5.9 (9.2)	8.7 (13.5)	
1.0kg (2.21b)	24 (75)	2.4 (16.8)	6.3 (9.8)	7.9 (12.3)	
8.9kg (19.61b)	18 (64)	1.6 (11.2)	2.4 (3.7)	3.5 (5.4)	
	25 (77)	2.0 (14.0)	1.7 (2.6)	3.1 (4.8)	
15kg (331b)	25 (77)	1.7 (11.9)	1.1 (1.7)	2.2 (3.4)	
Hens					
8.2kg (181b)	25 (77)	1.4 (9.8)	1.4 (7.2)	2.4 (3.7)	
4.4kg (9.71b)	18 (64)	1.4 (9.8)	2.4 (3.7)	3.3 (5.1)	
	26 (79)	3.8 (26.6)	1.0 (1.6)	3.5 (5.4)	

AIR TEMPERATURE AND LIVESTOCK PERFORMANCE

Optimum animal production is obtained within a range of thermal comfort. The environmental control system should aim to maintain building temperatures within specified limits if cold and heat stresses are to be avoided. Figure 3.2 illustrates the concept of animal production efficiency as it relates to temperature.

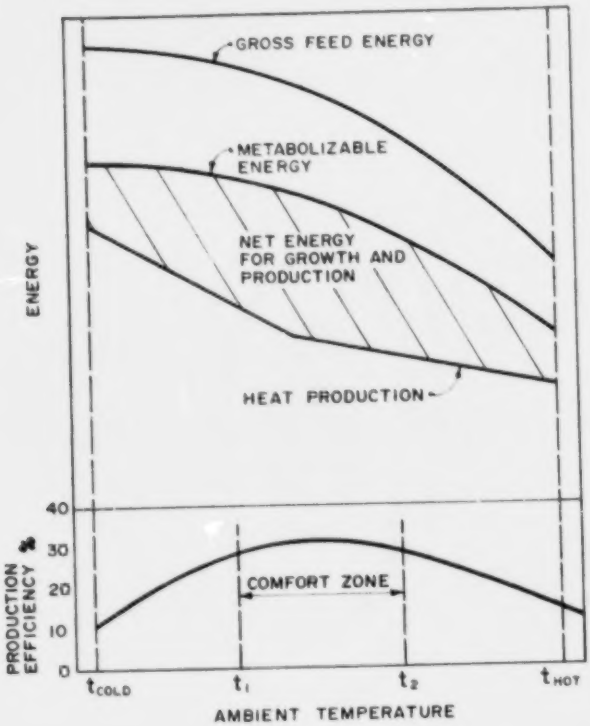


Figure 3.2 Proportion of food energy available for production as influenced by air temperature.

Figure 3.3 further illustrates animal response to temperature. The temperature range between A and A' is the zone of thermal comfort. In this range homeothermy is maintained by the animal without stress. These temperature limits are the same as t_1 and t_2 in figure 3.2

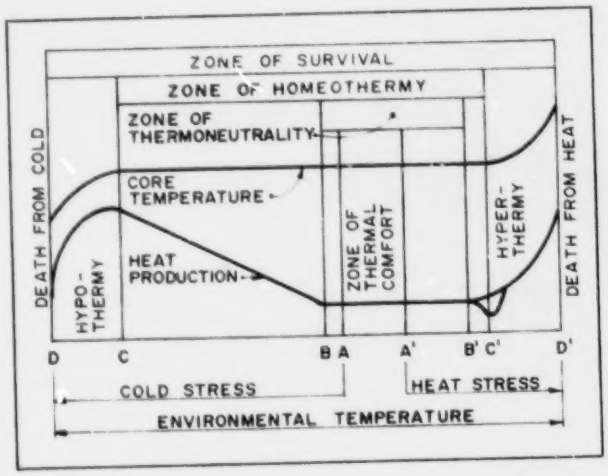


Figure 3.3 Homeothermic heat production and body temperature as affected by environmental temperature.

When environmental temperature drops below point A, blood vessels near the animal's skin will constrict and hair or feathers will become erect (called piloerection). The insulation value of the animal's fur increases since more air is held in the coat cover. Metabolic heat is conserved.

Below point B (the lower critical temperature), metabolic heat production must increase to maintain homeothermy. Increased feed intake is required to maintain animal production and the feed conversion efficiency decreases.

At point C homeothermy fails and body temperature drops. Heat production declines with decreased body temperature and death will ultimately result. Below point D death due to hypothermia is certain.

Time of exposure to extreme heat or cold is important. Short periods of stress between points C and D are survivable.

Above point A' the blood vessels near the skin will enlarge, and sweating and breathing rate (panting) will increase. These physiologial responses cause a greater rate of animal heat loss to the environment. Appetite will also be depressed to reduce body heat production. Animal productivity drops off.

Above point B' (the upper critical temperature) feed intake will diminish to near zero, evaporative mechanisms increase in intensity, and heat production (in some animals) drops off. Above point C' homeothermy fails and death from heat stress will ultimately result. Rising body temperature increases metabolic heat production while vapour loss from the skin and breathing rate are already at their maximum. Hyperthermia (heat fever) leads to death.

Figure 3.4 shows the effect of air temperature on the productivity of some classes of livestock and poultry. It is apparent from figure 3.4 that the peak productivity for laying hens, for example, occurs over a much narrower temperature range than for cows.

It is important to keep in mind that the optimum temperature zone varies with type and age of animal, changes in air velocity and, to a lesser extent, relative humidity.

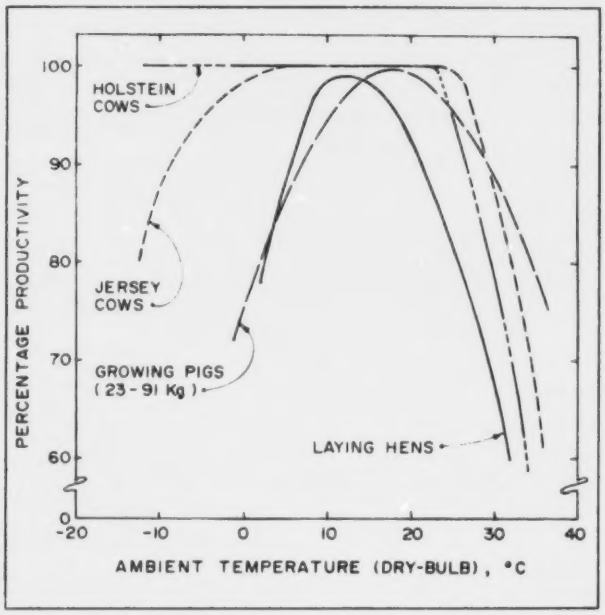


Figure 3.4 Percentage productivity of livestock and poultry as a function of air temperature.

Table 3.3 gives recommended air temperature ranges for livestock and poultry. Some caution is advisable when using this table. Optimum temperature tends to increase as air speed increases. This is due to increased convective heat loss from the animal. Table 3.3 recommends temperatures for conditions which are

typical of housing facilities for the class of animal listed. In order to achieve optimum production, producers may find it necessary to adjust air temperature above or below the optimum values shown.

TABLE 3.3

RECOMMENDED AIR TEMPERATURES

Livestock or Poultry	Temperature Optimum	°C (°F) Range
Swine		
Farrowing		
Sow	20 (68)	13-24 (55-75)
Litter - 1st day	35 (95)	35-40 (95-105)
Litter-newborn (<2.25kg or 5lb)	30 (86)	29-35 (84-95)
Litter-2.25kg (5lb) to weanling. Slowly lower to	24 (75)	18-25 (64-77)
Weanling 7 to 25kg (15 to 55lb)		
Batch housing		
start at	28 (82)	25-30 (77-86)
end at	23 (73)	20-25 (68-77)
Continuous housing (and use heat lamps)	24 (75)	21-27 (70-81)
Grower 25-60Kg (55-130lb)	20 (68)	14-24 (57-75)
Finisher 60 to 100kg (130-220lb)	16 (61)	14-24 (57-75)
Combined Grower Finisher		
25 to 100kg (55-220lb)	18 (64)	14-24 (57-75)
Breeding Room		
Group pen	18 (64)	16-20 (61-68)
Individual pen	20 (68)	16-20 (61-68)
Gestation Room		
Group Housed	17 (63)	16-25 (61-77)
Individual Stalls	20 (68)	20-25 (68-77)
Cattle		
Beef	13 (55)	5-21 (41-70)
Dairy	10 (50)	7-13 (45-55)
Calves	17 (63)	10-24 (50-75)
Veal		
Veal calves to 3 weeks	21 (70)	20-22 (68-72)
After 3 weeks	17 (63)	13-21 (55-70)

Livestock or Poultry	Temperature Optimum °C (°F)	Range °C (°F)
Sheep		
Newborn Lamb	30 (86)	29-30 (84-86)
Ewe	13 (55)	7-24 (45-75)
Maintenance 0 cm fleece	30 (86)	28-32 (82-90)
Maintenance 0.5 cm fleece	27 (80)	25-30 (77-86)
Maintenance 1.0 cm fleece	25 (77)	22-28 (72-82)
Maintenance 5.0 cm fleece	12 (54)	9-15 (48-59)
Maintenance 10.0 cm fleece	10 (50)	7-13 (45-55)
Full feed 0.5 cm fleece	20 (68)	18-22 (65-72)
Fasting 0.5 cm fleece	33 (91)	31-35 (88-95)
Horses		
Newborn foal	25 (77)	24-27 (75-80)
Horse	15 (55)	7-24 (45-75)
Rabbits	13 (55)	10-16 (50-60)
Poultry		
Laying hens	19 (66)	18-21 (65-70)
Broilers		
to 5 days (Brooding temperature)	35 (95)	32-35 (90-95)
5 days to 4 wks		
start at (slowly lower 2°C per week)	32 (90)	29-35 (84-95)
end at	21 (70)	18-24 (65-75)
After 4 weeks	20 (68)	16-21 (61-70)
Summer maximum temperature	29 (85)	
Turkeys		
0-5 days	35 (95)	32-35 (90-95)
5 days -2 wks	32 (90)	28-33 (82-91)
2-3 wks	29 (85)	27-31 (80-88)
3-4 wks	27 (80)	24-29 (75-85)
4-5 wks	24 (75)	21-17 (70-80)
5-6 wks	21 (70)	18-24 (65-75)
6 wks and older	18 (65)	15-20 (59-68)

AIR QUALITY

The concentration of contaminant gases in the animal environment can have an adverse effect on animal health and performance. A ventilation system should provide adequate air exchange to limit the concentration of gases, dust and disease pathogens.

Gaseous contaminants arise from manure decomposition, as byproducts of animal respiration and from fuel-fired heaters. Dust arises from feed, bedding, excreta, hair or feathers, and skin.

At the present time data on the effects of various airborne contaminants on animal health and performance are limited. Ongoing scientific research may better establish guidelines for maximum contaminant levels, however, in the interim, the following recommendations can be used for air quality standards.

Manure gases

The most prominent and deleterious gases released by decomposing manure are ammonia (NH_3), carbon dioxide (CO_2), methane (CH_4), and hydrogen sulfide (H_2S). Other volatile compounds, some of which are malodorous, are also produced but are not considered hazardous to animal health. Factors which influence, the type and nature of toxic manure gas emission are complex but include: animal species and age, temperature and pH of stored manure, manure storage time, and nature of the animal feed ration.

Ammonia (NH_3) is frequently encountered in high concentrations in swine and poultry confinement units. Ammonia is primarily an irritant and is rarely poisonous except in concentrations above 2000 ppm. Elevated ammonia levels have been linked to increased incidence of disease and respiratory aliments in animals. Ammonia has a sharp pungent odour and is highly soluble in water.

Hazardous level: 100 ppm
Recommended maximum: 20 ppm; 30 ppm for short periods
Recommended operating range: 5-15 ppm

Carbon dioxide (CO_2) is produced primarily by the animals rather than the manure. Unvented gas heaters can also contribute to CO_2 production. Normally CO_2 levels are not of particular concern to the ventilation designer since minimum ventilation is usually more than adequate for CO_2 removal. In mechanically ventilated buildings it is possible for CO_2 to reach dangerous levels during a power or equipment failure. CO_2 is not poisonous but is an asphyxiant; levels above 30,000 ppm lead to increased breathing. CO_2 is odourless.

Hazardous level: 60,000 ppm
Recommended maximum: 20,000 ppm
Recommended operating range: 500-5000 ppm

Methane (CH_4) is an asphyxiant but is not considered a direct health hazard at concentrations likely in even the most poorly ventilated barn. Its hazard lies in its flammable and explosive nature. A small spark or flame can trigger an explosion when the concentration exceeds 4 percent by volume in air. High concentrations of methane have been reported in the head space of enclosed manure pits, and explosions and fire damage in animal barns have been attributed to CH_4. Methane is odourless.

Hazardous level: 4% **FIRE HAZARD**
Recommended maximum: 0.5%

Hydrogen Sulphide (H_2S) is a highly toxic gas. Several deaths have resulted from exposure to H_2S. Agitation or mixing of manure sometimes causes lethal levels of hydrogen sulfide to be released. Adequate ventilation should be provided during removal of stored manure. H_2S has an extremely offensive "rotten egg" smell, but prolonged exposure fatigues the olfactory sense. Exposure at a concentration of 150 ppm can be fatal and rapid death ensues at 700-2000 ppm. Prolonged exposure by humans to H_2S levels between 5 and 10 ppm can lead to the respiratory disorder known as "dung lung".

Hazardous level: 150 ppm **FATAL**
Recommended maximum: 5 ppm
Recommended operating range: 0-1 ppm

Carbon Monoxide (CO) is a toxic gas produced by incomplete combustion of fuels in unvented heaters and brooders. It is not usually dangerous in properly ventilated buildings with heating units that are well maintained and installed according to codes.

Hazardous level: 1500 ppm **FATAL**
Recommended maximum: 35 ppm
Recommended operating range: 0-5 ppm

Dust

Dust contamination has been linked to respiratory disease. Micro-organisms are associated with dust and it is considered a significant disease vector. Unfortunately no firm limits on dust concentration have been established.

Factors which are related to dust level are: animal activity, stocking density, temperature, relative humidity, ventilation rate, air space per animal, feeding method, type of feed, and type of bedding.

Dust control may become an important feature of environmental control in the future, however at the present time methods of control are not well established. If a heavy accumulation of dust occurs, management practise may need

to be adjusted (see Trouble Shooting, Chapter 12). Ventilation rate has limited effect on dust level; feed and bedding type may need to be changed.

Relative Humidity

Relative humidity is related to air quality in terms of animal comfort; high relative humidities generally lead to a wet barn and poor evaporation. The ability of the animal to control body temperature by the mechanism of evaporative heat loss is limited when relative humidity is high. The "safe" operating range for relative humidity is between 40% and 80% for most animals.

Maximum permissible range: 40% to 80% RH
Recommended range: 50% to 75% RH.

CHAPTER 4 FANS

Fans are the driving force in any mechanical ventilation system. They are the "air pump" and as such serve two primary functions: (a) they create a pressure difference, and (b) they cause air to flow. In the application of livestock housing, the primary goal is to achieve substantial air flow rates with only a small air pressure difference between the inlet and outlet sides of the fan.

FAN TYPES

Fans may be divided into two types: **axial flow fans** and **centrifugal flow fans.**

Axial flow fans are generally used for livestock ventilation while centrifugal flow fans are used where a high pressure differential is required (e.g., grain drying). This discussion will be limited to propellor type, axial flow fans since these are the only kind used in livestock ventilation applications.

Propellor fans have two or more blades attached to a small central hub. The hub is either driven directly by the motor (direct drive), or by a belt and pulley mechanism (belt drive) which allows variation of fan speed. The blades are usually shaped from sheet metal. More expensive blades are cast and have cross-sectional shapes much like those of an airplane propellor.

Propellor fans are mounted in a circular ring or housing. Blade tip clearance is an important factor in fan performance. Small, uniform clearance between the blade tips and housing is desirable. This prevents air backflow around the propellor which would reduce fan efficiency.

Fan motors should be of the **totally enclosed** type for protection against dust and corrosion. Motors should, furthermore, be **rated for continuous duty use**, have built-in **overload protection** and for improved efficiency, never have a capacity greater than 25% over load requirements.

FAN RATINGS

In Canada there is no mandatory standard test for fan performance. This situation is unfortunate since manufacturer's performance data is often over-rated or misleading.

No fan is completely efficient; the best fans may reach as high as 80% total efficiency although most propellor fans are less than 40% efficient. Various factors limit a fan's performance and may include: propellor shape and design, housing design, presence of shutters and wind hoods, and motor performance. Static pressure difference, velocity pressure difference, static efficiency, mechanical efficiency and power input to the fan will also affect fan performance.

Because of the large number of variables affecting a fan's performance, it is desirable to know how the fan will react to the specific conditions in the barn. Shutters and hoods, for example, have a significant effect on fan performance. Fan manufacturers will often rate a fan without any of its accessories, leading to confusion and underdesign of ventilation systems.

In the United States, fan manufacturers have formed the voluntary trade association known as the **Air Movement and Control Association, AMCA.** The AMCA Certified Ratings Program provides uniform test procedures and ensures accuracy of published test data related to fans.

Fans bearing the AMCA sticker have been tested and more reliable performance data is available for them. Unless specifically stated, the AMCA test is on fans **without accessories.**

In Canada, while there is no mandatory test for fans, the **Prairie Agricultural Machinery Institute** periodically evaluates fan products in a comprehensive series of tests and publishes the results in the **PAMI Evaluation Report.** Where such information is available, it is reliable and should be used in place of manufacturer's data.

ENERGY EFFICIENCY RATINGS

With today's high energy cost, fan efficiency is becoming a much more important design consideration. Selecting energy efficient fans often represents a long term cost saving to the farmer. The higher cost of energy efficient fans will generally be repaid by decreased hydro bills.

An "energy efficient fan" is difficult to define. Not only are there various measures of efficiency but as technology continues to improve, an efficient fan today may well be considered inefficient tomorrow.

In general, large diameter fans are more efficient than small diameter ones. Efficiency tends to decrease as fan blade RPM increases and as motor size decreases. Efficiency is also a function of loading on the motor. Peak efficiencies usually

occur at 75% to 80% of full rated load.

Standardization of energy efficiency data has not occured and the designer must be careful of terms. One should be certain when comparing fan efficiencies to compare like terms. Another caution relates to reliability of information. It may be necessary to regard some manufacturers claims with caution until such time as a standard testing agency is established in Canada.

Important terms are defined below.

Fan or motor efficiency generally refers to the ratio of power output over the power input.

$$E = \frac{\text{Power out}}{\text{Power in}} \qquad \text{(Eq. 4.1)}$$

In the case of a motor, the power input is the amount of electricity used and the power output is the shaft horsepower. In the case of a fan, the power output is the **air horsepower delivered** (proportional to the volume flow rate x total static pressure difference).

Manufacturing variation refers to the fact that different motors and fans of the same design will not necessarily have the same efficiency. Manufacturing tolerances, variations in raw materials and quality control have an effect on efficiency. Fan efficiency for a given model of fan will consequently vary from unit to unit.

Nominal (or average expected) efficiency is the value which a large population of fans will have as an average when manufacturing variations are taken into account.

Minimum (or guaranteed) efficiency is the minimum efficiency value that a fan should have. It represents the bottom end of the range of efficiencies which a given model of fan will have owing to manufacturing variations.

Total fan efficiency refers to the entire fan and motor assembly. It often includes accessories such as shutters and hoods. Values for total fan efficiency will range up to about 50% for propellor fans. Total fan efficiency is useful as a comparison parameter between fans of similar size and capacity. Do not expect a 400 mm fan to have as good a total efficiency as a 1200 mm fan.

L/s/Watt or cfm/Watt is a measure on which fan manufacturers have begun to rate their fans. Since it states the flow rate delivered per unit of energy, it is a comprehensive measure of the fan's performance. It is a good standard of comparison between fans. Expect higher L/s/Watt values from larger

diameter, slower rpm fans. An energy efficient fan should have a L/s/Watt value of 5.6 (12 cfm/Watt) or higher. Values above 8.5 L/S/Watt (18 cfm/Watt) for fans smaller than 1000 mm diameter (40 in) should be regarded with some doubt. Given the present state of the technology, such claims are probably exaggerations on the part of the manufacturer.

The fan electrical bill is only a small part of the total farm energy budget. An energy efficient fan is just as wasteful of heat energy as a regular fan if it is operated incorrectly. In summary, it is a good idea to install energy efficient fans - but don't forget the other even more important energy conservation measures.

Fan payback calculation

A simple method of evaluating the economics of energy efficient fan selection is presented below. More sophisticated methods may also be used, but the following is useful as a first analysis.

Given

- 8000 hours/year operation
- energy cost of $0.05/kW-h
- "standard fan" $200
  1800 L/s (3800 cfm) fan with efficiency at full load = 3.8 L/s/W (8.0 cfm/W).
- "energy efficient fan" $300
  1800 L/s (3800 cfm) fan with efficiency at full load = 6.6 L/s/W (14.0 cfm/W).

$$\text{Payback} = \frac{\$ \text{ Premium}}{\$ \text{ Savings}}$$

$ Premium
= difference in cost of two fans

$ Savings
$$= (\text{electrical cost}) \times \frac{\text{hrs}}{\text{yr}} \times (\text{fan output}) \times \left[\frac{1}{Es} - \frac{1}{Ee}\right] \times \frac{1}{1000}$$

Ee = efficiency of energy efficient fan
Es = efficiency of standard fan

In this example, savings are:

$$= (0.05)(8000)(1800)\left(\frac{1}{3.8} - \frac{1}{6.6}\right)\left(\frac{1}{1000}\right)$$

= $80.38/yr

and payback period is:

$100/80.38 = 1.24 years

The cost of the energy efficient fan will be repaid by energy savings in 15 months.

At today's interest rates (14%), any payback time of less than 5 or 6 years indicates a net cost saving and the

energy efficient fan should be selected.

Fan Curves

Fan performance is often presented in the form of fan performance curves. Figure 4.1 depicts a typical fan performance curve and shows static pressure, efficiency, and brake horsepower, all plotted against air flow.

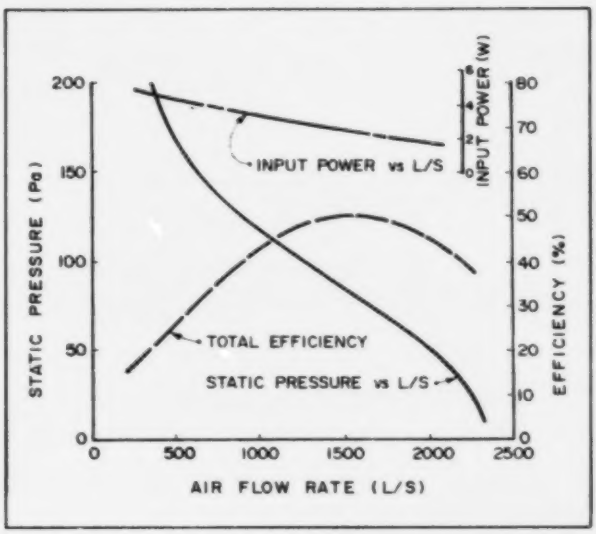


Figure 4.1 Fan performance curves for a typical axial flow fan.

The most important selection parameter for ventilating fans is airflow capacity at given static pressures.

Select fans on the basis of their airflow capacity when operating against a static pressure of at least 3 mm of water column (31 Pa or 1/8" of water column).

After satisfying airflow requirements, the noise level of the running fan may be an important consideration.

Noise level, measured in **decibels, dB(A)**, can be an irritation and a hazard to hearing and operator performance under continuous exposure. In general, noise level decreases in direct proportion to fan RPM. Fan noise levels above 70 dB(A) should be avoided. Ear protection above 85 dB(A) is definitely required.

FAN LAWS

Fan test results apply only to the specific conditions of testing. When fan speed is varied, the fan performance characteristics change. The "fan laws" provide a rough and ready estimate of fan performance characteristics when fan RPM or conditions are changed. The fan laws do not provide highly accurate predictions, however they are adequate for most applications when used within their limitations.

The fan laws can be used to predict air flow rate (Q), power (POW) and pressure difference (P) as functions of rotational speed (RPM), fan diameter (D), and air density (d). In the following relations the subscripts refer to states 1 and 2; state 1 refers to the fan for which data are known and state 2 refers to the fan for which characteristics are to be computed.

$$Q_2 = Q_1 \left[\frac{RPM_2}{RPM_1}\right]\left[\frac{D_2}{D_1}\right]^3 \qquad \text{(Eq. 4.2)}$$

$$POW_2 = POW_1 \left[\frac{RPM_2}{RPM_1}\right]^3 \left[\frac{D_2}{D_1}\right]^5 \left[\frac{d_2}{d_1}\right] \qquad \text{(Eq. 4.3)}$$

$$P_2 = P_1 \left[\frac{RPM_2}{RPM_1}\right]^2 \left[\frac{D_2}{D_1}\right]^2 \left[\frac{d_2}{d_1}\right] \qquad \text{(Eq. 4.4)}$$

Fan law restrictions are:

1. The laws are only applicable to fans which are homologous (having similar characteristics). Many axial fans are not homologous. Use of the fan laws to predict performance of fans of different model (or type) should be avoided or done with caution.

2. D_2 must be greater than D_1. (Don't use the laws to go from a larger to a smaller fan diameter.)

3. $(D_2/D_1) \leq 3$; $(RPM_2/RPM_1) \leq 3$; $(D_2/D_1) + (RPM_2/RPM_1) \leq 3$. The diameter and the speed ratio must individually be less than 3; their sum must also be less than 3.

FAN MOTORS

Single-speed

Minimum ventilation rates are often maintained by installing single-speed fans. Variation in ventilation rate can be obtained by installing a number of single speed fans. Single-speed fan motors are low in cost and available in a wide range of sizes.

Two-speed

Two-speed fan motors provide for a greater range of ventilation rates. Low speed capacity is usually between 1/2 and 2/3 of high speed capacity.

Variable-speed

Variable speed motors smoothly vary the fan speed from a minimum to a maximum. They provide the best ventilation rate versatility but have special problems of their own. Such motors are designed to be controlled by a temperature sensitive, solid-state variable speed controller. Since the load on the fan is so variable, the fan will rarely operate at its peak efficiency. Furthermore, at low speed settings the fan is highly susceptible to changing outside wind speed and direction. This is because the fan generates very little static pressure head at low speed settings. Consequently, problems with inlet air velocities can arise when insufficient negative suction pressure is generated.

Operating the motors at less than 20% of rated full speed can lead to motor overloading and burn out. Minimum speeds should be set at 20%-25% of full speed.

Some manufacturers claim satisfactory operation at 10% of full speed but this may lead to premature motor burn out and unreliable ventilation rates.

Another disadvantage of variable speed fans is that L/s/Watt ratings decrease when the fan is operating near its minimum speed.

RECOMMENDED VENTILATION RATES

Table 4.1 shows ventilation recommendations for several classes of livestock and poultry. As with all tables in this handbook, discretion must be used in the selection of animal class which best fits the design problem at hand.

TABLE 4.1

GENERAL VENTILATION RECOMMENDATIONS FOR LIVESTOCK AND POULTRY

TYPE OF LIVESTOCK OR POULTRY	TYPE OF HOUSING	MINIMUM WINTER RATE L/s-Unit(cfm/unit)	MAXIMUM SUMMER L/s-unit(cfm/unit)
DAIRY CATTLE			
450kg (1000lb) cow	Conventional fall to spring stabling. Ventilation by windows or doors during summer. Walls with RSI < 0.9.	10/animal (20)[20]	140/animal (300)[20]
450kg (1000lb) cow	Year-round well insulated RSI > 1.8 housing. Windowless or non-opening windows.	12/animal (25)[20]	210/animal (450)[20]
450kg (1000lb) cow	MILKING PARLOUR	--	190/stall (400)[17]
	MILK ROOMS		280/room (600)[17]
CALVES (Winter temp. controlled 7 to 15°C)			
Continuous housing: 50kg (110lb) average calf (1 month)	Year round housing in well-insulated barn	5/calf (10)[20]	35/calf (70)[20]
65kg (145lb) average calf (2 months)		5/calf (10)[20]	50/calf (100)[20]
Batch housing: 45kg (100lb) at start	Year round housing in well-insulated barn	5/calf (10)[20]	35/calf (70)[20]
135kg (300lb) at finish		7/calf (15)[20]	70/calf (150)[20]

Note: Superscripts denote reference number of source.

TYPE OF LIVESTOCK OR POULTRY	TYPE OF HOUSING	MINIMUM WINTER RATE L/s-Unit (cfm/unit)	MAXIMUM SUMMER RATE L/s-unit (cfm/unit)
BEEF 450kg (1000lb) cow	Walls with insulation RSI < 0.9. Ventilation by windows and doors during summer.	10/animal (20)[20]	140/animal (300)[20]
POULTRY **Laying hens**	Cages, bird density abouc 0.05m^2/bird	0.25/hen (0.5)[8]	3.5/hen (7.5)[8]
	Litter-high bird density 0.12m^2/bird	0.25/hen (0.5)[8]	2.4/hen (5.0)[8]
Breeder hens	Litter or mesh floor	0.25/hen (0.5)[8]	2.9/hen (6.0)[8]
Started pullets	1 or 2 deck cages	0.05-0.25/pullet[8] (0.1-0.5)	2.4/pullet (5.0)[8]
Broilers	Litter High bird density (≤ 0.09m^2/bird)	0.05-0.25/bird[8] (0.1-0.5)[8]	2.4/bird (5.0)[8]
Turkeys	Multi-purpose barns: rate per kg of bird (/lb bird)	0.125/kg (0.125)[8]	1.25/kg (1.25)[8]
	Turkey barns: rate per m^2 of barn (ft^2)	2.5/m^2 (0.5)[8]	25/m^2 (5)[8]
SWINE **Dry Sow**	Group pens	5.0/pig (10)[20]	95/pig (200)[20]
	Individual pens	5.0/pig (10)[20]	95/pig (200)[20]
Farrowing	Sow and litter	7.0/sow (15)[8]	132/sow (280)[8]
Weanling pig 7-25kg (15-55lb)	Batch housing	0.9/pig (2.0)[8]	15/pig (32)[8]
	Continuous housing	0.9/pig (2.0)[8]	15/pig (32)[8]
Grower Finishing Pig	Grower: 25-60kg (55-130lb)	2.4/pig (5.0)[31]	21/pig (45)[8]
	Finisher: 60-100kg (130-220lb)	3.3/pig (7.0)[31]	38/pig (80)[8]

TYPE OF LIVESTOCK OR POULTRY	TYPE OF HOUSING	MINIMUM WINTER RATE L/s-unit (cfm/unit)	MAXIMUM SUMMER RATE L/s-unit (cfm/unit)
	Combined: 25-100kg (55-220lb)	2.8/pig (6.0)[31]	28/pig (60)[8]
HORSES			
450kg horse (1000lb)	Year round insulated stabling, ventilated by windows and doors during summer.	10/horse (20)[8]	70/horse (140)[8]
SHEEP			
45kg (100lb) ewe	Walls with insulation RSI < 0.9 ventilated by windows and doors during summer.	1.0/ewe (2.0)[8]	7/ewe (15)[8]
RABBITS			
Doe and litter	Cage with density of 14kg/cage (30lb/cage). Rate in L/s-kg(cfm/lb)	0.125/animal[8] (0.125)	1.25/kg (1.25)[8]
	Lower density	0.075 (0.075)[8]	0.90/kg (0.90)[8]
CHINCHILLAS	Year round, insulated housing (in cages). Mature. L/s-animal(cfm/animal)	0.05/animal[8] (0.125)	1.2/animal (2.50)[8]

FAN SIZING AND SELECTION

1. Determine the minimum and maximum ventilation rates for the room by the use of table 4.1, General Ventilation Recommendations for Livestock and Poultry.

2. **Know what kind of controls you are planning to use before ordering your fans.** Consult with the control system dealer or manufacturer. Plan the ventilation system as a single, integrated system, not as a collection of parts which you hope will give good results.

3. Assume a static pressure of 3 mm (1/8 in) negative pressure to determine fan output. Do not use "free air delivery" figures. These will seriously undersize fan capacity once installed in the barn.

4. Do not size fans on the basis of their diameter. Use the best data available for rated fan performance.

5. Select fan motors which are **totally enclosed** (for dust protection), which are rated for **continuous duty use**, and which have **thermal overload protection**.

6. Select fans on the basis of their efficiency. Energy efficient fans will almost always save money in the long term. Large, slower rpm fans generally have higher efficiencies than small fans.

7. Use the winter, minimum rate to select fans which will run continuously.

8. Use the summer, maximum rate to design a system using different sized single-speed fans, two-speed fans or variable speed fans.

9. Provide for at least three steps in the ventilation rate from minimum to maximum.

Installation and Maintenance

Fan blade position in the housing is extremely important. Fan efficiency drops off substantially if blade placement is incorrect. Figure 4.2 shows the correct placement of the blade in the housing.

Some motor mounts are of a poor design and the fan and motor can slip out of position. The farmer should be informed of this potential problem and shown how to correct it.

Location of fans in the building is not particularly important. See **Chapter 5, Inlets**, for more details on fan location. If possible, fans should be 200 mm (8 in) below continuous air inlets and obviously

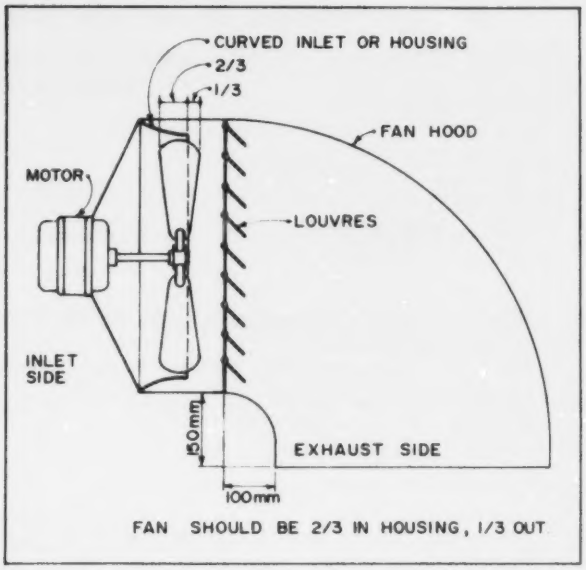


Figure 4.2 Correct placement of fan blade in housing and proper hood design.

should not be placed directly downstream of the inlet. In general, two guidelines for fan placement are used:

1. Avoid placing fan in situations where short circuiting of air supply may occur (2400 mm or 8 ft from any potential air inlet, e.g., window, door, etc., where the incoming airstream is directed at the fan).

2. It is preferable to balance fans on both sides of the building. That is, where continuous slot inlets are used that discharge toward both sides of the building, have an equal fan capacity on both side walls. This configuration provides better symmetry of airflow from the inlets. (See chapter 5.)

Put proper shutters and hoods on all exhaust fans. Wind effects can seriously reduce the output of a fan if no hood is used. A 30 km/h wind blowing directly toward the discharge airflow from an unprotected exhaust fan will reduce the exhaust capacity by approximately 90%!

Shutters (or louvres) prevent back-drafts when fans are stopped. Hoods help to reduce the effects of wind on fan performance. They also prevent shutter freeze up during cold weather. It is a good idea to use a common hood for both cycling and continuous operation fans. The continuous operation fan provides protection against freeze-up for all fans under the common hood.

The position of the hoods and their design are important. Hoods should turn down a full 90°, and extend at least 150 mm (6 in) below the fan opening and 100 mm (4 in) out from the outer wall to prevent air back-flush (see figure 4.2).

Ensure that adequate wiring is installed. Inadequate wiring can lead to low-voltage fan-motor overload. All wiring should meet electrical code standards and all equipment should have CSA approval.

Fans should be maintained once a month. The maintenance list should include all

of the following:

1. Check blade position with respect to housing (2/3 in 1/3 out).

2. Clean blades, guards, shutters, motors and housing - the whole unit. **Dirty fans are a ventilation system's biggest enemy.**

3. Check motor bearings for wear and fan rotation for balance. Replace if necessary.

4. Check and adjust belt tension on all belt-drive fans.

5. Lubricate motors, bearings and shutters.

Dust and dirt build-up can reduce fan output by more than 50%. Keep fans clean.

Fan Catalogue

The fan catalogue which accompanies this handbook provides selection data for various fans. It is as complete as current information allows. Information has been gathered from the most reliable sources available, however in most cases this is the manufacturer. Since no mandatory testing program for fans exists in Canada, all of the comments and warnings made earlier regarding reliability of fan performance data should apply.

Parameters cited should be self explanatory, however they are stated only in Imperial units. Convenient conversions for the use of this catalogue are:

1 cfm = 0.472 L/s

1 cfm/Watt = 0.472 L/s-W

1/8" s.p. = 3 mm s.p. = 31 Pa

1 in = 25.4 mm

1 hp = 745.7 W

CHAPTER 5 INLETS

Inlet design is probably the most important part of the ventilation system. A ventilation system which is perfect in all other ways will perform poorly if inlet design is inadequate. Inlets present a difficult design problem. Methods for design of inlets have not yet been perfected but it is possible to install a good inlet with designs available today.

It is the inlet, not the fan, which provides good air movement throughout the barn. The fan is only the air pump. The functions of an air inlet are:

- To bring sufficient quantities of fresh air into the room;
- To provide directional control of the air flow;
- To maintain inlet air-stream velocity so that good air mixing between fresh and room air occurs;
- To distribute the air evenly throughout the room;
- To prevent drafts.

VENTILATION PHILOSOPHIES

Ventilation by Displacement

A displacement-type ventilation system is one in which the air enters the building along one wall surface and passes as an intact wave through the room to exhaust vents on the opposite wall. As the "wave" or "slug" of air proceeds through the room airspace, stale or contaminated air is displaced on a 1:1 volumetric basis.

If a real system could be designed to employ the displacement concept it would indeed be an excellent system. Ventilation efficiency would be maximized since one roomfull of supply air would be sufficient to displace (and remove) one room-full of contaminated air. Moreover, if the air-flow direction was vertical, then cross-contamination between animals by airborne disease vectors would be eliminated.

Unfortunately, any deviation from the ideal renders the ventilation by displacement concept invalid. The assumption inherent in the theory is that no mixing of old and new air occurs. Air flow must be unobstructed and of a velocity small

enough to be entirely non-turbulent. In a real barn this is difficult to achieve. Even a small amount of animal movement destroys the smooth laminar air-flow pattern and practical penning arrangements pose obvious obstructions to air flow. Mixing of fresh and room air is unavoidable.

Practical experience with displacement ventilation systems has shown that they have too many problems. Ongoing research may eliminate these problems, but at the present time displacement ventilation systems are not recommended.

Ventilation by dilution (mixing)

A dilution type ventilation system is one in which turbulent mixing of fresh and room air occurs. The objective is to provide perfect mixing by means of appropriately designed and placed air inlets. High velocity inlet air jets cause mixing of incoming air with room air. Gases and water vapour are diluted by the fresh incoming air.

In the case of perfect mixing, complete removal of air contaminants is never possible. After one room-full of fresh air exchange, 35% of the original room air remains. Four air exchanges yield a 99% reduction in initial room air, but 100% air contaminant removal is never achieved.

Unlike displacement systems, turbulent mixing of supply and room air is induced purposely to dilute contaminated air everywhere within the room at the same rate.

The advantage of dilution ventilation over displacement ventilation may be simply summarized: neither perfect plug flow nor perfect mixing are possible to achieve in a real ventilation system, but it is at least possible to **approximate** perfect mixing while it is nearly impossible to approximate plug flow. A displacement system merely ends up as a poorly designed dilution system.

INLET PRINCIPLES

In most livestock ventilation systems, the energy to promote air mixing within the airspace is provided largely by the jet action of the air at the supply inlet.

A cold air jet issuing horizontally along the ceiling through a ceiling level slot will either fall or remain horizontal depending on its momentum and density relative to the room air. To prevent cold drafts, an attempt is made to keep the jet attached to the ceiling where mixing of the cold air with warm room air will occur.

There are numerous problems with such jets. Generally, the recommendation is

to have an air velocity of 5 +- 1 m/s (1000 +- 200 ft/sec) at the inlet. Unfortunately, the problem is more complex than the simple specification of air jet velocity. Consult the Bibliography for more information on the theoretical considerations of jet dynamics (References 3, 4, 9, 13 and 24).

Relationships for Ventilation Air Jets

There are two basic types of air jets: free jets and wall jets. The free jet issues out of an orifice or slot and is not adjacent to or influenced by solid boundaries. The wall jet issues out of an inlet with one edge coinciding with a wall or ceiling.

The air jet has a characteristic velocity profile as shown in Figure 5.1. Kinetic (velocity) energy of the jet is dissipated as room air is entrained in its flow cross-section. The action of entrainment induces secondary air movement in the room.

Vena Contracta and Coefficient of Discharge

Figure 5.1 also illustrates the concept of **vena contracta**. When air issues out of a slot of a given width, say 30 mm, the actual area of flow is **less** than the width of the slot, for example, 20 mm. This phenomenon is due to the fact that the air must curl into the opening; the effective area of flow is diminished by the sharp bend the air must make as it passes through the slot.

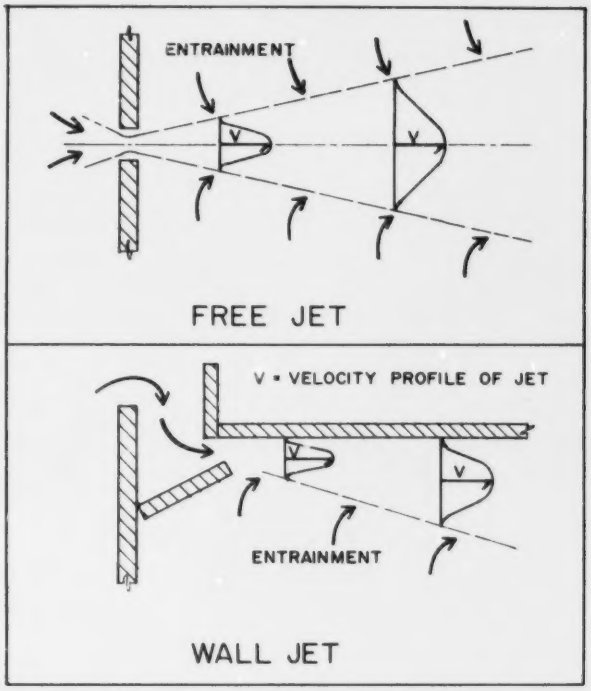


Figure 5.1 Air jets showing vena contracta, entrainment and velocity profiles.

The precise design of the slot or inlet will dictate the amount of vena contracta and other effects due to air friction. The **coefficient of discharge, C_d**, can be experimentally determined for any given inlet. It may be inserted into the **continuity equation** (equation 5.1a) as a correction factor.

$Q = VA$ (Eq. 5.1a)

$Q_a = V_a A_a$; $C_d = 1.0$

or

$Q = VAC_d$ (Eq. 5.1b)

$Q_a = V_a A_a C_d$

Q_a = Volumetric flow rate, m^3/s or cfm
A_a = Area of opening, m^2 or ft^2
V_a = Velocity of air jet, m/s or ft/min
C_d = coefficient of discharge

Values of C_d may be obtained from the manufacturers of most inlets. If in doubt, a value of C_d = 0.7 is a good starting value for most inlets and avoids the incorrect assumption of no vena contracta or friction.

AIR MIXING PATTERNS

The objective of inlet design is to induce desired air flow circulation patterns in the barn. Most of the design considerations are based wholly on practical experience since air flow patterns are difficult to predict theoretically.

Air circulation patterns are influenced by all of the following:

- Inlet design and location;
- Air jet velocity, temperature and density;
- Ventilation rate;
- Animal movement and heat and moisture production;
- Room layout and design: pen walls or other obstructions which block or deflect air flow;
- To a limited degree, fan location.

Air flow never follows a straight line from inlet to exhaust; air flow better resembles a series of generous circular or rotary motions. Figure 5.2 shows that air flow patterns are composed of **primary** and **secondary (or induced) flows.** Obstructions such as pen walls lead to increased secondary air movement. Smaller projections such as strip lights, corrugated steel ceilings (or even the animals themselves) cause further secondary air movement.

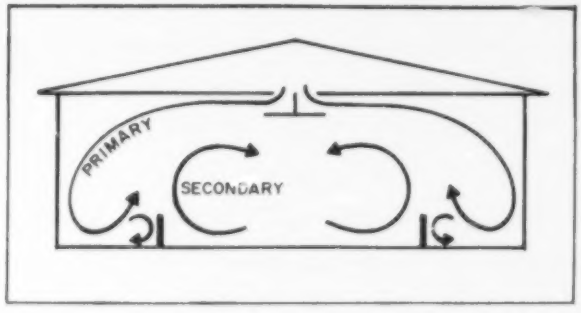


Figure 5.2 Primary and secondary air circulation in the barn.

The layout of the room must consider air circulation. Room layout should locate pens in a manner best suited to the likely air flow patterns. Figures 5.3 and 5.4 show desirable airflow patterns in two typical swine barn cross-sections. The objective is to direct the air so that it reaches the dunging area first. This results in a slightly less comfortable dunging area and better dunging habits.

Because jet throw is limited, wide barns

must have more inlets than narrow barns. For barns less than 5000 mm (16 ft) wide place continuous ceiling level inlets along a single sidewall. For barns up to 12,000 mm (40 ft) wide, place continuous ceiling level inlets along both sidewalls or use a centre-mounted interior ceiling inlet. For wider buildings add one or more interior ceiling inlets. Figure 5.5 illustrates the principle. In order to achieve desired air circulation patterns, the exact configuration of the interior air inlets may be different from those shown.

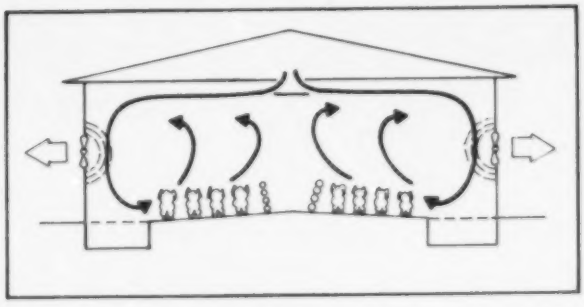


Figure 5.3 Desirable air flow in a swine barn with centre feed alley.

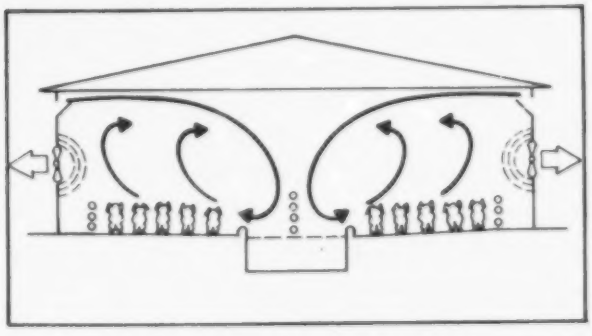


Figure 5.4 Desirable air flow in a swine barn with central dunging passage.

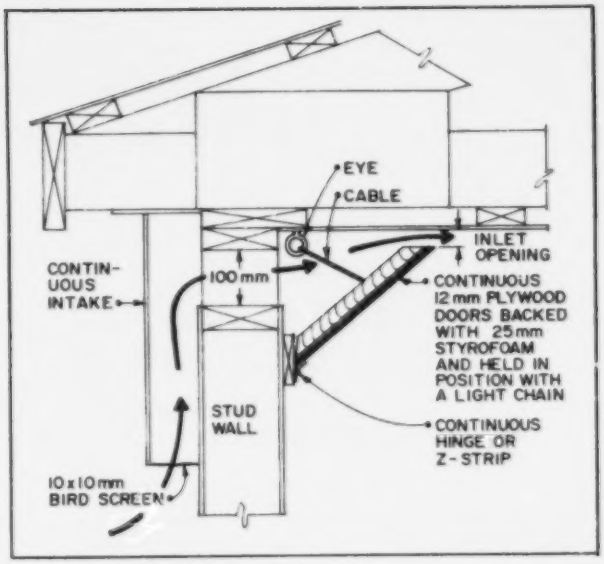

EYE
CABLE
INLET OPENING
CONTIN-UOUS INTAKE
100 mm
CONTINUOUS 12 mm PLYWOOD DOORS BACKED WITH 25 mm STYROFOAM AND HELD IN POSITION WITH A LIGHT CHAIN
STUD WALL
CONTINUOUS HINGE OR Z-STRIP
10 x 10 mm BIRD SCREEN

Figure 5.5 Recommended inlet configuration for barns of varying width.

INLET DESIGN

Sizing and Adjustment

After determining minimum and maximum ventilation rates and the general barn layout, the inlets must be sized. Adjustment for the range of ventilation rates is essential. Continuous slot inlets are strongly recommended and sizing formulas are presented below.

SI (Eq. 5.4)

$$I = \frac{Q}{5LC_d}$$ (for 5 m/s jet)

Imperial (Eq. 5.5)

$$I = \frac{Q}{1000LC_d}$$ (for 1000 ft/min jet)

I = Inlet opening, m or ft.

Q = Air flow rate, m^3/s or cfm

L = Length of inlet, m or ft

C_d = Coefficient of discharge
= 0.7 if no other value is available

The above formulae give the required inlet opening, however adjustment is necessary to be able to maintain a 5 m/s jet velocity for the full range of ventilation rates. Use the maximum ventilation rate to determine the maximum inlet size required. The inlet should be designed to handle the minimum ventilation rate by closing nearly shut if necessary.

Design Criterion

The inlet design should satisfy the following constraints:

- Direct air along ceiling.
- Control jet velocity within 4 to 6 m/s and airflow volume between the minimum and maximum ventilation rate.
- Inlet should be easily adjustable by winch or automatic control.
- The inlet panels should be insulated to prevent condensation.
- The inlet should be continuous over the entire length of the building.
- It should be simple to build and inexpensive.
- The inlet should not be obstructed in any way; **the ceiling must be smooth.** No lights, water pipes, wiring, or other obstructions should disrupt the air jet issuing from the inlet.

Inlets and Fan Placement

The location of the fan has very little bearing on the performance of the inlets or on air circulation patterns. The importance of fan location is often overrated by ventilation designers. Fans may be spaced along the side walls of the barn or be grouped in banks to simplify electrical wiring.

A continuous slot running over the top of fan banks will not lead to severe air short circuiting. Some designers suggest leaving at least 2,400 mm (8 ft) of solid wall between banks of fans and continuous slot inlets. This is not necessary. Limited short circuiting will occur when the slot is continuous over the top of the fan, but the advantages of such a design outweigh this disadvantage. A well adjusted inlet will jet the air away from the fans.

If the following recommendations are used, fans will perform well (provided the fans are properly designed and sized).

1. Use continuous inlets over the top of the fans but leave 200 mm (8 in) of wall between the bottom of the inlet and the top of the fan opening.

2. Space the fan evenly along side wall or group them in banks. In either event there should be no more than 18,000 mm (60 ft) between fans or fan groups.

3. If possible balance the fans for symmetry on the side wall and on both sides of the barn. The examples below explain.

   - If 4 banks of fans are required, put 2 on each side wall of the barn. If the wall is 30,000 mm long, centre the two fan banks so that they are 15,000 mm apart and 7,500 mm from each end wall.

   - If the barn is narrow (less than 5,000 mm wide) and the inlet is on a single side wall, then either place the fans under the inlet or on the opposite side wall.

   - If the stepped control circuit cuts different fans in at different times, make sure the system will still be balanced. Don't have 4 fans operating on one wall and 1 fan operating on the other. Or don't have 3 fans operating at the far end of the barn and none at the other.

4. Avoid fan and inlet configurations which will obviously lead to short circuiting. An open window, door, manure pit vent or other opening next to a fan will cause short circuiting, as will an inlet discharging directly towards a fan (when the fan is less than 2,400 mm (8 ft) in a direct line from the inlet).

POSSIBLE INLET DESIGNS

Side Air Inlet

The side air inlet shown in figure 5.6 is a good basic design. The opening is controlled by means of a wire cable which passes through an eye in the ceiling. By pulling on the wire (with a winch or motor) the inlet is closed. A weight on the other end of the control wire pulls the inlet open when the line is slacked off.

Notice that the inlet of figure 5.6 is insulated and that there are no strips, jamb plates or other obstructions on the ceiling. The intake on the outside wall of the barn is continuous and provides protection against wind, much like a fan hood.

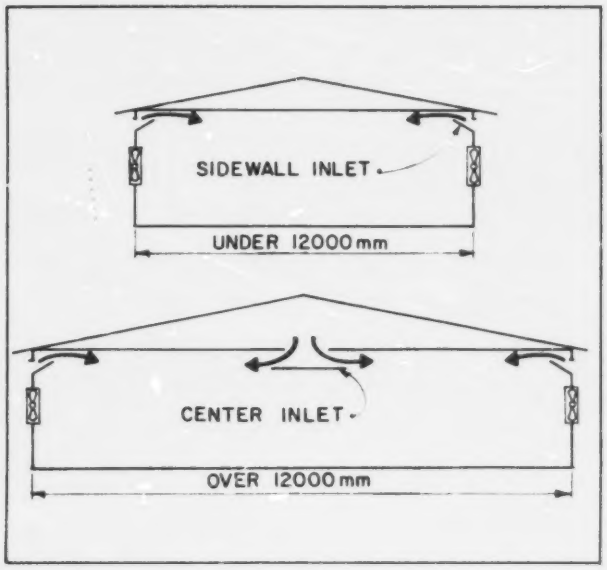


Figure 5.6 Example side air inlet design.

Centre Air Inlet

The centre air inlet shown in Figure 5.7 is a Canada Plan Service design (plan no. 9711). The opening is controlled by a winch and cable mechanism which can be operated manually or by a motor.

Curtain Inlet

The curtain inlet shown in Figure 5.8 is a type of self-adjusting inlet. As air flow increases, the fabric curtain flexes and opens wider. The curtain is weighted with wooden dowels or lead weights.

One of the problems with curtain inlets is that the material ages and deteriorates. Dust collection on the fabric and fabric breakdown often makes the inlet brittle. The performance characteristics of the inlet change with time.

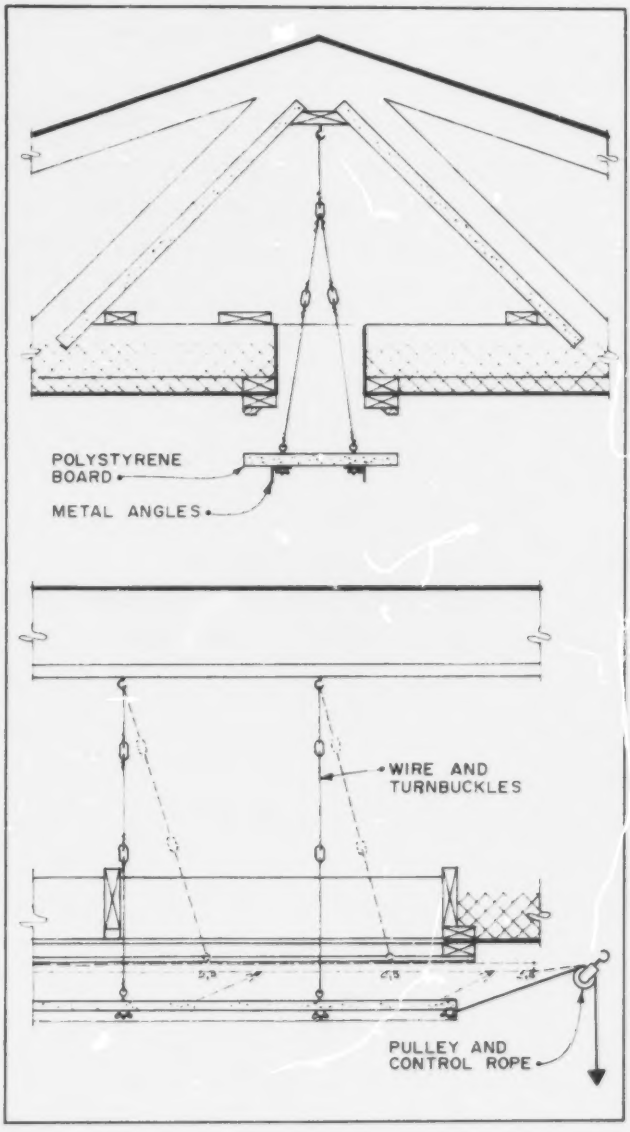


Figure 5.7 Canada Plan Service centre air inlet design.

The advantage of a curtain inlet is that it responds instantly to changes in static pressure. Proper adjustment of weights to ensure the development of a

good jet velocity and static pressure difference is essential. Replacing the old and deteriorated fabric once every few years is also important.

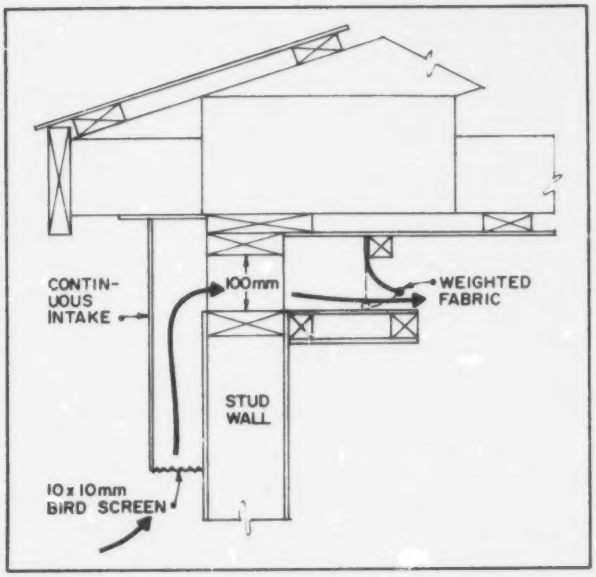


Figure 5.8 Example curtain air inlet design.

Gravity Inlet

Figure 5.9 shows another type of self-adjusting inlet which uses counter weights instead of flexible fabric. The gravity inlet is basically a modified side air inlet. The counterbalance holds the inlet shut when the fans are off; as air begins to flow the static pressure pulls the inlet open.

The advantage of this system is in its instant response to changing ventilation rate. The disadvantages are in the tedious set up and adjustment of the weights to obtain the right air jet and static pressure difference.

Both curtain and gravity inlets must be protected against wind. Wind action can cause either type of inlet to be blown open.

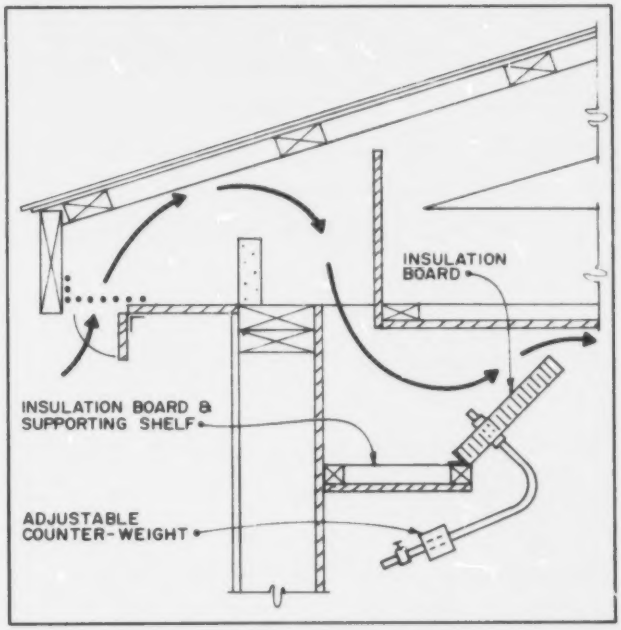


Figure 5.9 Gravity inlet with counter weights.

FAN JET OR POLY TUBE SYSTEMS
(see chapter 8)

Automatic Inlet Controls

Various electromechanical inlet control devices are available in the market-place. These use a guage to measure static pressure difference between the inside and outside of the building

(usually a "photohelic guage"). Two set points are adjustable on the guage: a minimum and a maximum static pressure difference. When the static pressure drops, the guage activates a motor which closes the inlets. Similarly, the motor opens the inlets when pressure rises.

Such control systems generally work well and eliminate the labor and inaccuracy of manual inlet adjustment. They are, however, subject to erratic performance by wind induced pressure differences. The operator can switch to manual operation during periods of high wind.

Inlet Operation

Where no automatic control is used, **the inlets should be adjusted at least twice a day:** once in the morning and once in the evening. Whenever fan output changes the inlets should be adjusted. Naturally this is not possible where a manual system is used.

The formulae given in equations 5.4 and 5.5 may be used to assist in inlet adjustment. A static pressure guage and smoke test should be used to determine optimum setting.

The optimum static pressure guage reading will vary from barn to barn depending on differences in inlet configuration and on how air-tight the building is. The objective is to achieve a jet velocity of 5 m/s +- 1 m/s (1000 ft/min +- 200 ft/min) at the inlet. This velocity will usually occur for most modern confinement barns when the inlet is adjusted to achieve about 2 mm (.08 in) of water column static pressure difference. Higher jet velocities may produce better circulation patterns during winter operation.

Inlets must be kept clean and in good working order. "Sticky" inlet doors should be repaired promptly.

COMMERCIAL SYSTEMS

Two examples of commercial inlet designs are depicted below. Both examples are presented for illustration purposes only. Presentation of these products should not be construed as endorsement by either the author or the B.C. Ministry of Agriculture and Food.

Chore-Time

The Chore-Time product uses box type inlets with sliding panels to vary inlet area. Inlet hoods are installed on exterior walls to provide protection against wind effects.

Adjustment of the Chore-Time system is by cable and winch. Panels are connected by cable to a hand or motor driven winch. All of the inlets are adjusted simul-

taneously when the cable is adjusted. Figure 5.10 is a sketch of some Chore-Time system components.

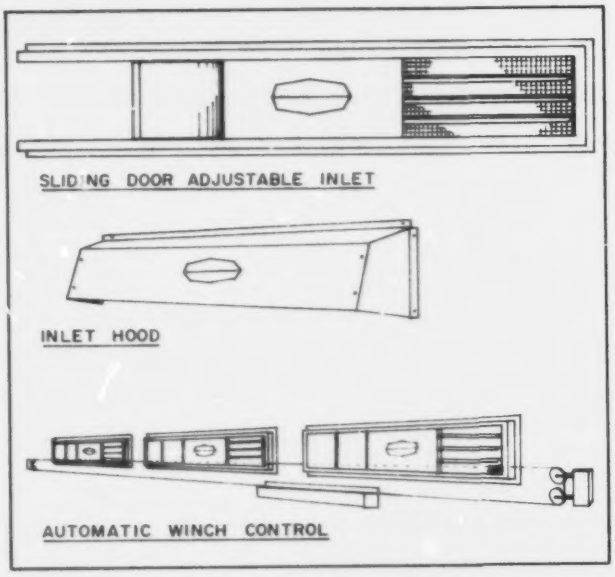


Figure 5.10 Chore-Time inlet control system.

Martin-Air

The Martin-Air product line employs a powered-inlet/powered-exhaust concept which affords the potential for air mixing and recirculation.

Several different models of the Martin-Air system are available besides the wall mounted unit illustrated in Figure 5.11. Centre-ceiling mounted, and offset-ceiling mounted systems are used to induce different air circulation patterns.

The Martin-Air systems utilize a deflecting baffle or diffuser hood on the inlet to direct air evenly through the room. The fans operate continuously while the damper in the mixing chamber provides for different levels of air mixing.

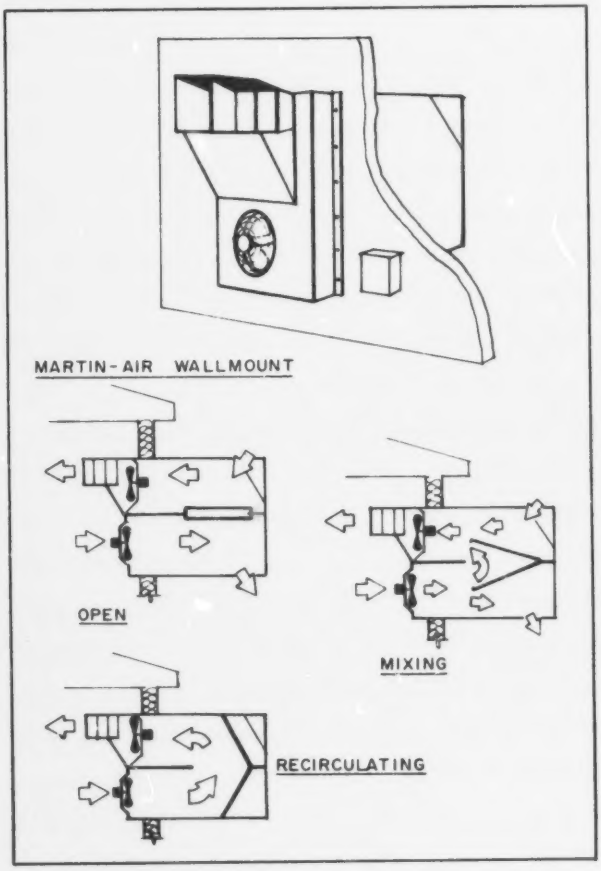


Figure 5.11 Martin-Air powered inlet.

CHAPTER 6 CONTROLS

This chapter deals with methods for controlling the operation of the fans, inlets and heaters in a controlled animal environment. The types and methods of control systems are numerous and rapidly evolving. The micro-processor (computer) promises to improve the accuracy, reliability and versatility of environmental control systems.

This chapter is divided into two sections. The first section (**Conventional technology**) deals with control systems presently in common use. The second section (**Developing technology**) deals with the control system which will likely become common in the next few years.

CONVENTIONAL TECHNOLOGY

Conventional heating and ventilation control systems for agriculture are based on two measurements (and sometimes only one): dry-bulb temperature and building static pressure. Occasionally a humidistat will be added to the system for measurement and control of relative humidity.

Thermostats

Thermostats are the most common type of mechanism for control of ventilation equipment. A thermostat is composed of a sensing element (measures temperature) and an electrical switch (tripped by the sensing element). There are three types of sensing elements used in thermostats: bi-metallic, vapour filled, and liquid filled.

Bi-metallic thermostats use a two-sided bi-metallic coiled strip as the sensing element (also called the power element). The two types of metal on either side of the metallic strip expand or contract as temperature changes. Since different metals have differing coefficients of thermal expansion, the bi-metallic strip tends to coil or uncoil with changing temperature.

As the coil distorts, it trips the electrical switch. The switch may be of the floating contact or the mercury type. Both types of switch are prone to "chattering" which rapidly deteriorates contacts and propogates line voltage spikes which cause problems with associated electrical equipment.

Bi-metallic thermostats are common for household applications but are not well

suited to agricultural use. Corrosive gases deteriorate the bi-metallic strip, calibration is imprecise, and differential (defined below) is generally too large.

Vapour filled thermostats are often used for control of agricultural heating and ventilation equipment. The sensing element contains a vapour-liquid mixture. As temperature rises the vapour expands and exerts a pressure on the switch, causing it to be tripped.

The switching force in a vapour filled sensing element is small, hence such thermostats are not susceptible to decalibration due to temperature extremes or distortion of the bulb. Vapour filled elements can be designed for excellent sensitivity to small temperature changes. Lag times (defined below) are short, but differential may be larger than desired.

Liquid filled thermostats are similar to the vapour filled type except that the bulb and bellows is filled with a liquid (usually mercury) and no gas or vapour phase is present. The switching force is provided by the expanding or contracting liquid. Liquid filled sensing elements have a wide temperature range, low differential and relatively long lag time.

Mechanical damage of the bulb and capillary tube will lead to de-calibration or failure of the instrument. Exposure to temperatures outside of the temperature range of the thermostat will also cause decalibration and loss of accuracy. Liquid filled thermostats are often used in agriculture and, if of good design and construction, are suitable.

Accuracy (sometimes called "drift") of the thermostat is a measure of how close the measured temperature is to the actual temperature. Accuracies of +- 1°C (+- 2°F) are common and generally acceptable but such information may be difficult to obtain from the manufacturer. **It pays to buy quality thermostats:** poor accuracy is often the cause of incorrect fan staging.

Differential of the thermostat is the difference in temperature between switching points on a temperature rise or decline. For example, if a thermostat is set at 20°C and has a 2°C differential it will turn the heater on at 19°C and off at 21°C.

A certain amount of temperature differential is desireable since extremely small differencials lead to excessive cycling of equipment. This is hard on switching contacts and equipment. Differentials around 1°C (2°F) are ideal while differentials of 3°C (5°F) or larger should be avoided.

Lag is the time duration between a change in actual temperature and the thermostat's

measurement of the change. Liquid filled thermostats have longer lag times because the sensing element has a relatively high mass (it takes longer to heat a large mass than a small one).

Lag can lead to a "cycling effect" or set point overshoot. When a heating system is first turned on, for example, the lag time may be large and there will be an overshoot past the first set point. In normal operation lag time is not usually large enough to cause serious cycling problems.

Range is the range of temperatures over which a thermostat will perform according to specifications.

Thermostat switches are of several types. In general, line voltage, snap action, totally enclosed micro-switches are most suitable for use in a livestock environment. Floating contacts, as mentioned earlier, are prone to chattering and should not be used. A new trend is to solid state switches which are more expensive but extremely well suited to agricultural applications. Their extra cost is justified by their improved reliability.

It is important to check the switch voltage and current capacity of the switch against the system loads. **Never exceed the capacity of a thermostat switch.** 24 volt control circuits are often used in conjunction with relay switches to provide more precise control of equipment. 24 volt systems are reliable providing floating contact switches are not used.

Where 3-phase systems are encountered it is necessary to use relay switches for equipment control since almost all thermostats are of the single-pole type.

Single-pole single throw or single-pole double throw switches are used in thermostats. The single throw switch activates a single-speed fan whereas the double throw switch controls two-speed fans.

Solid state electronic phase control is most common for variable speed thermostats. As temperature changes, a thermistor in the control circuit causes the voltage delivered to the fan motor to vary. This controls fan speed.

Variable speed thermostats usually have a **control band** having a **band width** of between 3°C and 6°C. The set point is in the middle of the control band and corresponds to fan operation at 50% of maximum speed. At the high end of the band the fan runs at full speed and at the low end of the band the fan runs at its pre-set minimum speed.

Variable speed thermostats have a minimum speed screw (found on the back of RANCO

thermostats) to prevent low-voltage fan motor overloading. This should be set at a minimum of 20-25% of full rated fan speed to prevent motor burn out. It is wise when installing variable speed thermostats to allow access to this control screw. Taking out one of the rear knockouts on the electrical box will allow access to the set screw found on RANCO products.

Thermostat Control Design and Installation

The premise of thermostat control is that level of ventilation is determined by building temperature; as temperature increases so does ventilation rate for temperature control.

The number of possible fan/thermostat control configurations for any given system is very large. The designer should use common sense when considering control logic and must be satisfied that every possible mode of operation is acceptable and desirable.

The first logical step in control design is "what equipment do I have to run". Identify the heaters, fans, cooling systems, etc. that are to be run by thermostat control.

Next, identify the alarm and safety features desired. For example battery powered electric power failure alarms are common, but you may wish to add a telephone dialing alarm which automatically telephones pre-programmed numbers if the temperature in the barn is too high or too low for any reason.

Third, identify all other related equipment: static pressure guages, inlet control motors, shutter and damper motors, humidistat sensors and controls, or any other part of the environmental control system. The objective is to have the entire system working interdependently, so consider all of the components and how they are to operate.

Fourth, determine the fan staging sequences desired for the fans selected. Special control thermostats are necessary for variable speed fans and control circuits will usually be separate from single or two-speed fans in the system. **It is necessary to have complete information on fan current and voltage requirements in order to prevent overloading thermostat switches.**

The next critical step is the minimum ventilation rate logic. It may be desirable to have some ability to adjust minimum ventilation rate for different stocking densities. If single speed fans are used for minimum ventilation rate they do not need to be connected to a thermostat; single speed fans should be connected directly to the main power

supply on separate or parallel power circuits. Since maintaining the minimum ventilation rate is critically important, separate circuits prevent complete system breakdown should a breaker switch be tripped.

Where variable or two-speed fans provide minimum ventilation rate it is necessary to use thermostats. Again, it is wise to connect minimum rate fans to separate circuits and thermostats.

Factors which become important for set-up of thermostats are as follows:

1. Number of thermostats
   - Because of thermostat accuracy (drift), setting and adjustment for accurate fan staging becomes more difficult with increasing numbers of thermostats - from a management point of view it is desirable to have few thermostats.

2. Thermostat location
   - The most reliable measurement of air temperature is at animal level, however the risk of mechanical damage to the instruments forces placement above animal level.

   - Placement of thermostats in stagnant or drafty areas results in unrepresentative room temperatures and poor control.

   - It is more effective to install an air circulation system than it is to put a thermostat beside each fan - it is a mistake to have 10 thermostats controlling 10 fans.

   - Exposure to radiant heat (sunshine, heaters) leads to erroneous operation as does attachment to a cold exterior wall.

Stepped control of fans is achieved by thermostat activation of additional fans as temperature rises. Two examples of the control logic follow.

Example 1

Broiler barn 10,600 birds
Minimum ventilation rate 530 L/s
Maximum ventilation rate 25,400 L/s
Desired temperature 29°C (Brooding temperature)
See example design problem number 6.

- Select two 1400 L/s variable-speed fans c/w single modulator for minimum rate, with minimum speed set to 20% of maximum for brooding period. Adjust minimum speed setting upwards as birds grow.

- Select four 3000 L/s variable speed fans, c/w two modulators in series with two thermostats set at 30°C minimum set point. Adjust set point

downwards as birds grow.

- Select two 6000 L/s single-speed fans, single-pole single-throw thermostats set at 31°C and 32°C, separate circuits.

- Mode of operation: 560 L/s to 2,800 L/s continuous.
  Above 30°C: 5,200 L/s to 14,800 L/s variable in two steps.
  Above 32°C: 20,800 L/s to 26,800 in two steps.

Example 2

Swine finisher barn 392 pigs
Minimum ventilation rate 1290 L/s
Maximum ventilation rate 14,900 L/s
Desired temperature 16°C
See example design problem number 4.

- Select two two-speed (650 L/s / 1300 L/s) fans, single-pole double-throw thermostats set at 10°C 1st set point - 16°C 2nd set point, separate circuits.

- Select two single-speed 2500 L/s fans, single-pole single-throw thermostats set at 18°C.

- Select two single speed 4500 L/s fans, single-pole single throw thermostats set at 20°C.

- Mode of operation: 1300 L/s continuous.
  Above 16°C: 2600 L/s
  Above 18°C: 7600 L/s
  Above 20°C: 16,600 L/s

In order to avoid faulty fan staging it is usually necessary to have at least 1°C difference between set points on the various thermostats. This fact is due to the accuracy or drift of the thermostats. When two identical fans are used for one ventilation step, it is advantageous to put the fans on the same thermostat; this ensures that both fans will go on and off at the same time thus eliminating some of the difficulties in faulty fan staging.

Thermostats are also used for control of heaters and cooling systems. Heaters may need to be electrically interlocked with fan control thermostats to prevent simultaneous operation of both fans and heaters (see chapter 11, Heating). Cooling systems generally do not conflict with fan operation since they do not operate unless temperatures are high and all fans are already running at full capacity.

Timers

10-minute cycle timers are often installed on fan/thermostat circuits and **are the source of many ventilation problems.** 10-minute cycle timers are used to override the thermostat and achieve

minimum ventilation rate with a fan that has a larger than minimum rate output. The timer causes the fan to operate intermittently.

It is much better to size fans to run continuously since intermittent fan operation almost invariably leads to periodic fluctuations in room temperatures and humidities. **10-minute cycle timers should be removed and the oversized fans replaced with smaller, properly sized fans.**

Humidistats

In some cases humidistats are used for minimum rate ventilation for moisture removal. They are extremely difficult to keep properly calibrated when used in a dusty environment. Accuracy, even when clean, cannot be expected to be better than +-5%. Humidistats of any kind are probably best used for **monitoring** of humidity level, rather than control, since they are not reliable.

Humidistats commonly use a hair filament which elongates or contracts depending on relative humidity. Electric **hygrometers** use a principle of changing electric resistance in a thin-surface electrolytic salt solution. Both types of instrument are subject to decalibration and failure by dust and chemical action.

Typical Thermostat Control Circuits

Figures 6.1 and 6.2 demonstrate standard one and two-speed fan/thermostat circuits.

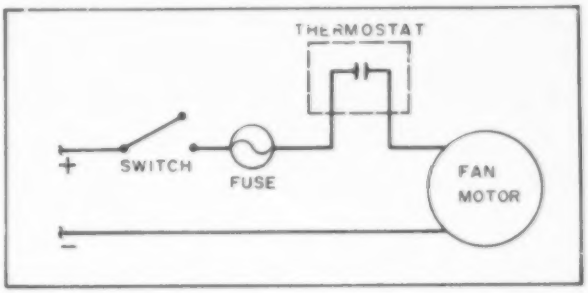


Figure 6.1 Simple one-speed fan control thermostat.

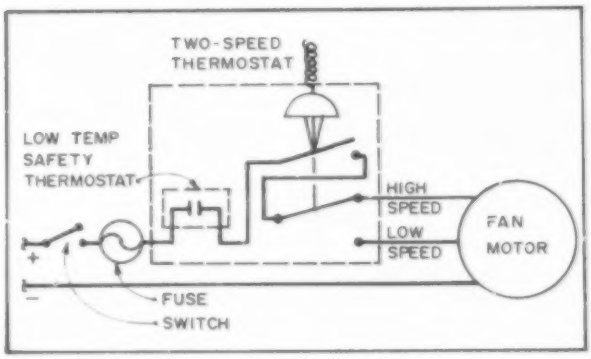


Figure 6.2 Simple two-speed fan control thermostat.

Figure 6.3 shows a simple battery operated thermostat alarm circuit. It is easy to install in an existing facility because it does not require splicing into existing fan circuits. Its disadvantage is that it does not warn of power failure.

Figure 6.4 shows a more sophisticated alarm system which is activated by both power failure and temperature. The relay switch closes if power at the fan is lost.

Where more than one fan is included in the alarm circuit, a relay is required for each fan.

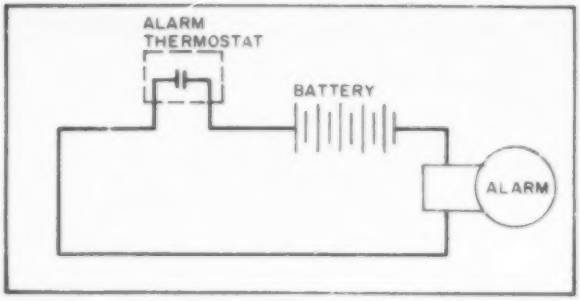


Figure 6.3 Battery powered temperature activated alarm.

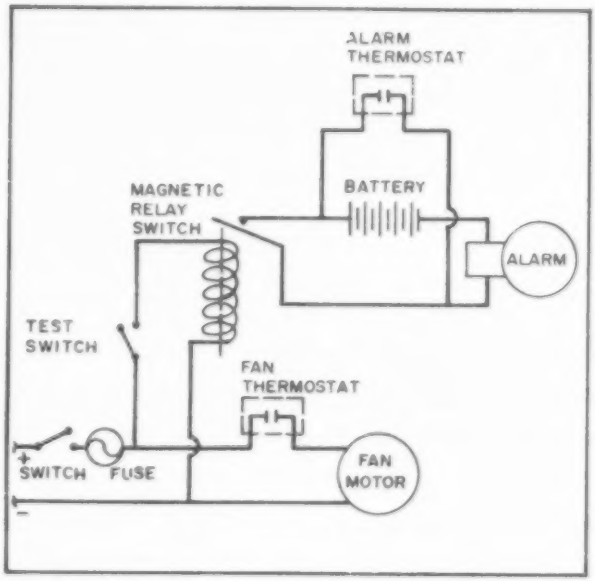


Figure 6.4 Alarm circuit activated by power failure or temperature.

DEVELOPING TECHNOLOGY

Computer technology and the revolution in the electronics industry is certain to improve environmental control for livestock and poultry. Two developments make this technology possible:

1. The microchip

2. Remote sensing equipment

The microchip provides the farmer with an inexpensive computer which is capable of analysing information from various sensing devices and directing equipment to respond appropriately. Such an automatic control system will have the capacity to optimize environmental conditions, while minimizing energy and management costs.

Developments in the area of micro-processor environment control are proceeding rapidly. At the present time there are two problems which are slowing progress. The first of these is a lack of suitable sensing devices to measure wet-bulb temperature and various contaminating gases and dust. The second problem is a lack of appropriate software (computer programs) for livestock environment control. Both of these problems are certain to be overcome as the industry develops.

Micro-processor Control

The heart of the system is known as a **micro-processor** and is composed of one or more wafer-thin electronic components called **microchips**. The microchip is

capable of storing information and a series of instructions or computer programs. Complex instructions can be programmed to operate all of the various parts of the barn heating and ventilation system.

The micro-processor is capable of doing more than just turning fans on and operating and closing inlet vents - it can also collect valuable information on system and animal performance for use by farm managers. What follows is a brief list of possible functions a farm computer system could perform.

1. Collect measured data and record it continually.
   - Barn temperature
   - Outside temperature
   - Barn relative humidity
   - Outside relative humidity
   - Barn static pressure and inlet air velocity
   - Outside wind direction and velocity
   - Barn gas and dust concentrations
   - Feed consumption of animals or birds
   - Water consumption of animals or birds
   - Average weight of animals or birds
   - Fan and heater electrical consumption

2. Based on collected data, control equipment.
   - Control operation of all fans
   - Control operation of heating/cooling system
   - Control inlet settings
   - Control feed delivery system
   - Control water delivery system
   - Control waste handling system

3. Provide important analysis of data for farm management.
   - Graphical summaries of all barn and outside climatic data
   - Calculate daily feed and water consumption
   - Calculate feed conversion ratios
   - Compute feed allotments for animal management and ordering purposes
   - Calculate energy consumption in the barn
   - Predict motor failure of equipment by monitoring current draw

4. Signal malfunctions by alarm or telephone.

5. With additional information and programs:
   - Record and manage individual animal performance
   - Manage animal breeding

There are several different design concepts for micro-processor control of ventilation systems. These depend on where and in what manner the following

parts of the system are located.

1. Remote sensing equipment (temperature, etc.)
2. Data input (keyboard for changing desired room temperature, etc.)
3. Micro-processor or Micro-computer
4. Control algorithms (equipment control programs)
5. Monitoring and analysis algorithms (analysis of data programs)
6. Display (video terminal, printer)
7. Equipment control circuitry (e.g., inlet vent control)

Remote sensing equipment sends information to the computer on temperature, static pressure and other measurements so that the computer can make decisions about equipment operation.

The **data input keyboard** allows the farmer to tell the computer what the temperature, static pressure and other conditions in the barn should be so that the computer can compare values from the sensors with desired values. If the system uses a micro-computer like an IBM personal home computer, then the data input board may be like a typewriter. Or the data input board may consist of a few buttons like a calculator keyboard and could be located just about anywhere.

The **Micro-processor** is the brain of the system and does not necessarily have to be located in the barn. Sometimes the micro-processor is split into parts, one part of it in the barn and another part at the farmer's home or office.

The micro-processor contains both the **control algorithms** and the **monitoring and analysis algorithms**. These are computer programs which tell the computer what exact steps it must follow to analyze and organize the information it receives from the remote sensors and the data input device. These programs also tell the computer how to operate the ventilation equipment.

The **display system**, be it a television screen, printer, or calculator-type display, allows the computer to tell the farmer what the computer is doing. Graphs, lists, daily high and low temperatures and other information can be seen and recorded for the farmer's use.

The **equipment control circuitry** (signal conditioning devices) interface the signal from the computer with controls on the fans, heaters, inlets and other equipment. Each piece of equipment requires a suitable signal conditioning

interface between itself and the computer. These devices are generally located in close proximity to the ventilation equipment.

Examples of a simple system and a more sophisticated (and expensive) system are presented below. Both examples are fictional but based on available technology.

Simple In-Barn Micro-processor

- 8000 bird brooder barn
- Inside thermistors
- Outside thermistors
- Outside wind speed and direction indicator
- Humidistat inside
- Building static pressure manometer
- Roof mounted alarm

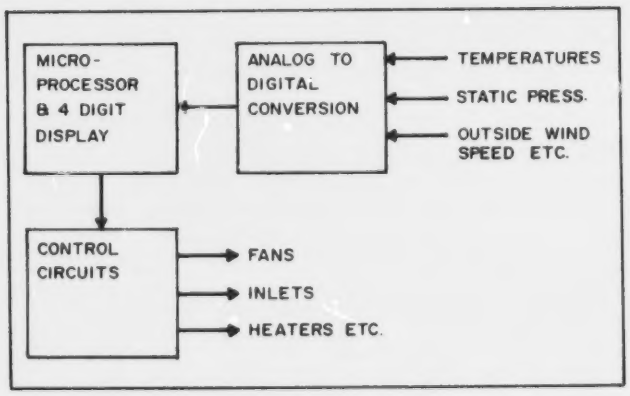


Figure 6.5 Single barn micro-processor control schematic.

This micro-processor control system is self-contained in the barn. It consists of the above list of sensors which are connected by wires to a small box about 150 x 500 x 500 mm (6" x 20" x 20"). On the box cover there are 4 buttons and a small liquid crystal display. This allows the farmer to input desired temperatures, building static pressure and relative humidity.

The micro-processor averages the temperature readings from the thermistors in the barn to obtain the average room temperature. If heat is required, the brooders are turned on by a relay switch and an electric gas valve. The fans are operated to maintain temperature and relative humidity within the desired ranges. The computer is programmed to prevent simultaneous operation of the brooders and step 2 exhaust fans. No energy is needlessly wasted.

The micro-processor opens and closes the inlets to achieve the desired static pressure in the building. If outside wind speed is greater than 25 km/h the inlets are moved to a preprogrammed position and an alarm on the barn roof is activated. Similarly, if temperature goes out of the range specified by the farmer, or if there is a power failure, the alarm is sounded.

The whole package is neat and simple. The barn's environment is well controlled but the farmer does not have the benefit

of detailed graphs, printouts or other useful information. The system would cost about $3,500 installed.

Advanced Multi-Barn Micro-computer System

- 6 barn broiler operation
- Inside thermistors
- Inside humidistat
- Static pressure manometer
- CO_2 sampler
- Ammonia sampler
- Inlet air velometer
- Feed delivery monitor
- Water delivery monitor
- Bird weighing device
- Telephone dialing alarm

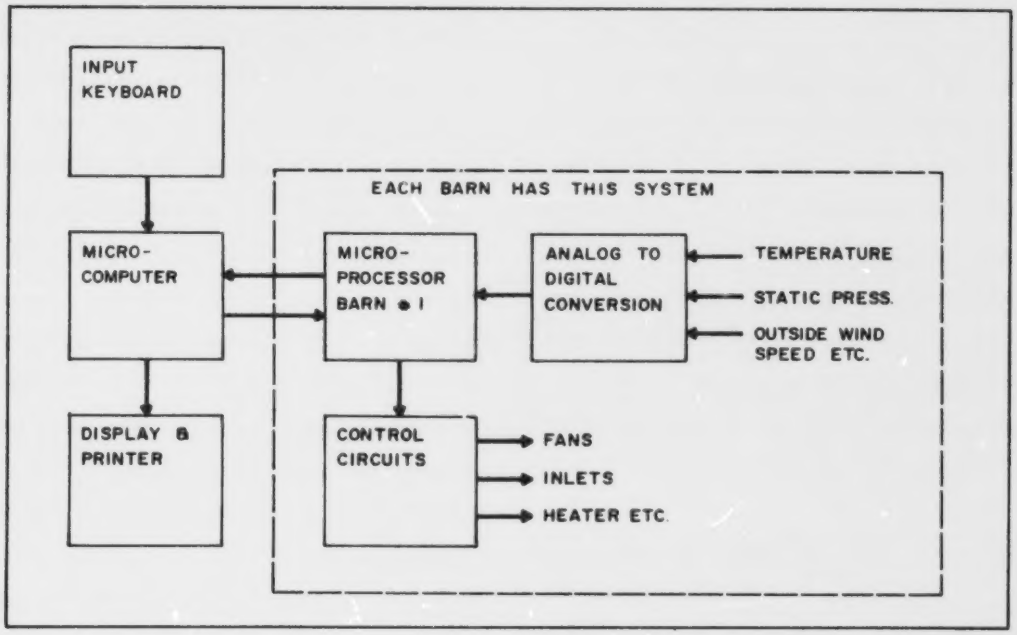


Figure 6.6 Multi-barn micro-processor control system schematic. A central computer is connected to a satellite micro-processor in each barn.

This micro-computer control system has a main computer system in the farm house with a video display, printer and key-board. Each barn also has a micro-processor which controls individual barn environments. The micro-processor sends information to the main computer for analysis and display on the video terminal. Each barn is controlled according to the farmers specifications for temperature, relative humidity, carbon dioxide and ammonia. All equipment control is automatic. Sensor readings of the environment in each barn are shown on the video terminal. Graphs, charts and listings of past conditions are all available to the farmer on the video and

printer. An automatic telephone dialing alarm warns of any problems with equipment or barn environment.

Management information on flock performance is also prepared by the computer. Water consumption, feed consumption, daily weight gain and feed conversion ratio along with predicted feed requirements and other information is available at the farmer's request.

The increased capability of this system is justified by the size of the operation (6 barns). Individual systems such as the one described earlier could also be used, but the cost per barn would probably be comparable. At the time of writing such a system would likely cost between $20,000-25,000 to install.

Limitations of the Micro-processor

The first thing to realize about a micro-processor control system is that it can't perform miracles; **a micro-processor will not change a poor ventilation design into a good one.** To take full advantage of micro-processor control, first rate barn construction is required. But a micro-processor can operate a given system to its maximum potential. This is because the micro-processor makes logic and control decisions on a continuous basis which no human manager could match even if he attended to manual controls 24 hours a day.

The major limitations of a micro-processor control system are inherent in its design and installation. These can be divided into 3 sections:

1. Hardware design and installation.

2. Quality of the controlling logic - the software.

3. The relationship between 1. and 2.

The **hardware** consists of the remote sensors, the computer and its accessories. The **software** consists of the control logic and command programs. Software is extremely important - it is the area which requires the most development and improvement. Finally, the relationship between the hardware and the software is critical since the desired operation will occur only if all of the parts of the system communicate with each other consistently and correctly.

Weaknesses of the hardware system may include:

- Inadequate or improper installation of remote sensors (e.g. improper placement of thermistors resulting in unrepresentative temperature measurement).

- Poor quality, innaccurate sensing instruments.

- Poor interfacing of sensors with micro-processor (e.g. analog signals require conversion to digital form - bad wiring, cheap electronic components etc. will cause serious problems).

- Expensive maintenance requirements. The system should be designed for ease of repair. Manual override of the micro-processor should be built into the design.

- Ability of equipment to withstand the very harsh (to electrical components) environmental conditions of most animal environments: dust, corrosive airborne chemicals and moisture.

- Computing power - is the system large enough, fast enough and versatile enough to control all of the various pieces of equipment?

- Not enough input and output channels to monitor and control all of the desired equipment.

Weaknesses of the software may include:

- Incorrect or oversimplified control logic which does not make the right decisions to control equiment properly.

- "Bugs" in the program which cause the computer to make mistakes or "crash" (fail completely and require restarting).

- The computer may not be "user friendly" (be easy to understand or operate). Some computer languages are more sophisticated than others and are therefore easier for a non-computer expert to use.

- The software may not be powerful enough - it may not perform all the functions and data analysis that the farmer needs or finds useful.

Weaknesses in the relationship between hardware and software may include:

- A mismatch of capability: the software may be much more sophisticated and powerful than the rest of the system or the opposite may be the case.

- The software may not be compatible with the hardware.
  - Signals from sensors may not be understandable to the software;
  - Signals sent by the software may not be understandable to the control equipment.

Remote Sensors

Thermistor

In micro-processor applications the thermostat is replaced by a **thermistor**. This simple device measures temperature by electrical means and is extremely small and compact.

Thermistors are made from semiconducting materials whose electrical resistance decreases with a rise in temperature. There are many different kinds and classes of thermistors for different applications and temperature ranges. Their advantage is that they are easily interfaced with the computer, accurate, and have virtually no lag time.

The other functions of the thermostat (set points, line voltage switch, etc.) are taken over by the micro-processor. The micro-processor compares the thermistor temperature with the desired temperature and makes the necessary adjustments to fans, heaters and inlets.

Humidistats

Electric hygrometers are generally used for their compatibility and ease of interfacing with the micro-processor. Hygrometers are particularly sensitive to dust and corrosive gases in the barn environment.

Wet-bulb temperature

Relative humidity may be measured directly with a hygrometer or indirectly by measuring either wet-bulb or dew-point temperature. The RH is related to the wet and dry bulb temperatures by psychrometric relationships and can be determined if both are known. An alternative to the humidistat is therefore automatic measurement of either wet-bulb or dew-point temperature.

Unfortunately no inexpensive devices currently exist which can measure either t_{dp} or t_{wb} in a dusty and dirty environment. This technology may be developed in the near future as the need for it is widely recognized.

Gas samplers

Control of ventilation rate on the basis of contaminant gas concentration is a new possibility of micro-processor control technology. The problem is again a question of reliable and inexpensive gas sensors for CO_2, CO, NH_3, CH_4 and H_2S. While there do not appear to be any such suitable sensors at the present time, this technology should be watched closely for future developments.

Pressure Manometer

There are a range of devices available which measure total building static pressure. Inlet adjustment is usually controlled by attempting to hold static pressure constant at some value (usually somewhere near 2 mm of water column). The optimum static pressure will vary from barn to barn depending on how tight the construction is and the specifics of the inlet design.

CHAPTER 7 NATURAL VENTILATION

In this chapter, two types of buildings that can utilize natural ventilation systems are discussed - cold environment barns and warm environment barns. Both types of building require careful planning for the successful operation of the ventilation system.

COLD ENVIRONMENT HOUSING

Cold environment housing is common in mild climates. Almost all beef cattle, sheep and dairy cattle barns are cold confinement buildings in the South Coastal Region of B.C. The objective of this type of housing is to allow the interior temperature to follow (within only 2 - 3°C) the exterior or ambient temperature. No insulation is required to retain heat in the building and condensation is controlled by allowing plenty of fresh air to flow through the building, thus reducing the inside to outside temperature differential.

Side-Wall Vents

Natural air movement is provided through openings in the side and end walls of the barn. Ventilation by displacement of air occurs whenever the wind blows. For this reason, exposure of the barn to prevailing winds is important for effective natural ventilation. The barn should be oriented so that the ridge line of the barn is perpendicular to the direction of prevailing winds.

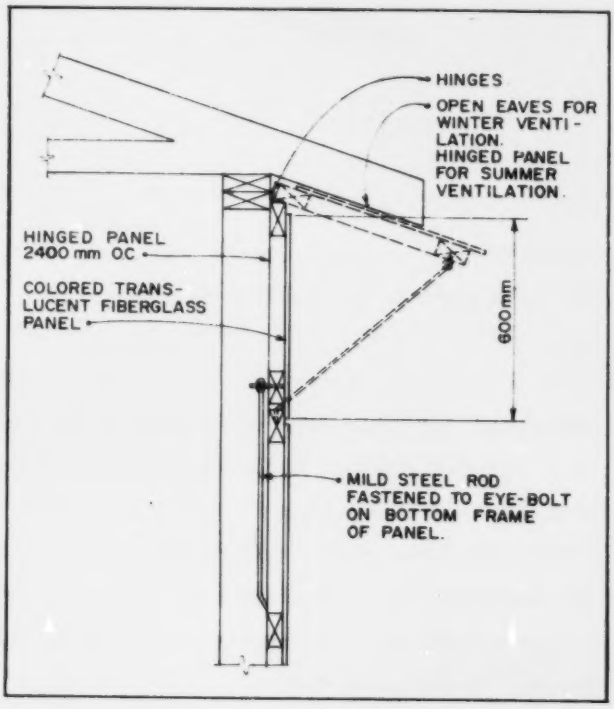

HINGES
OPEN EAVES FOR WINTER VENTILATION.
HINGED PANEL FOR SUMMER VENTILATION.
HINGED PANEL 2400 mm OC
COLORED TRANSLUCENT FIBERGLASS PANEL
600 mm
MILD STEEL ROD FASTENED TO EYE-BOLT ON BOTTOM FRAME OF PANEL.

Figure 7.1 Side-wall panel opening for natural ventilation.

The rule of thumb for **size of wall opening is 25 mm per 3000 mm (1" per 10 ft.) of building width along both side walls.** The opening should be continuous for the entire length of the wall and

should be located on both walls. Therefore, for a 12000 mm wide building there should be a 100 mm deep, continuous opening along both side walls.

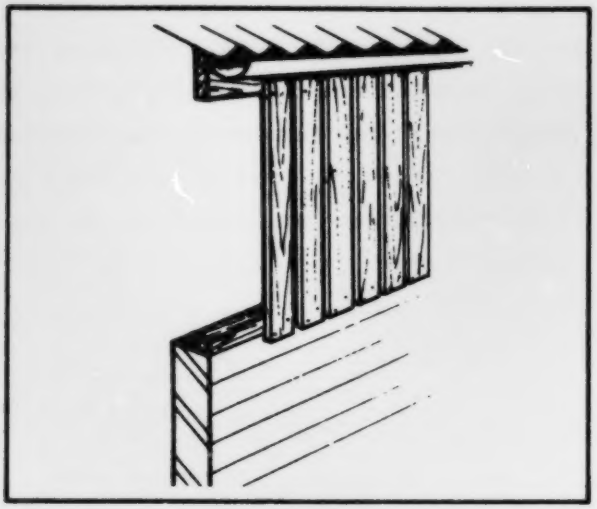


Figure 7.2 Spaced panel boarding used as a windbreak to prevent drafts.

The required openings can usually be provided by allowing the space between the top plate or beam and the roofing material to stay open. With some construction methods and for very wide buildings, it is necessary to build-in the required opening at the top of the side wall.

With few exceptions, eave openings should remain fully open throughout the year. In situations where blowing snow becomes a problem, a hinged baffle can be provided to partially close the eave inlet during the winter storms.

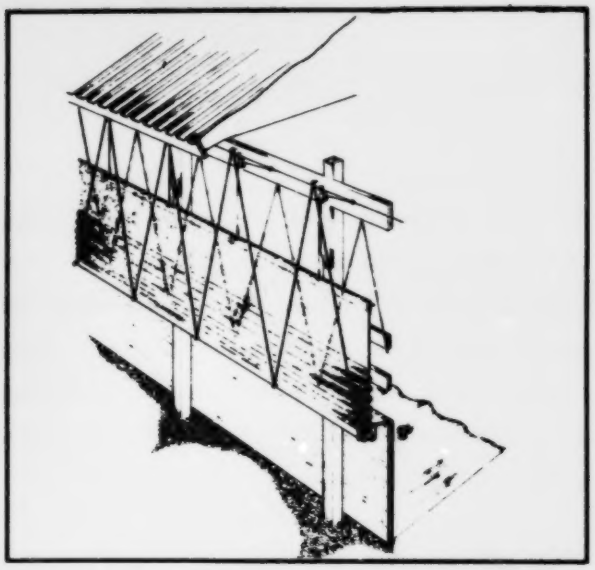


Figure 7.3 Plastic curtain wall opening.

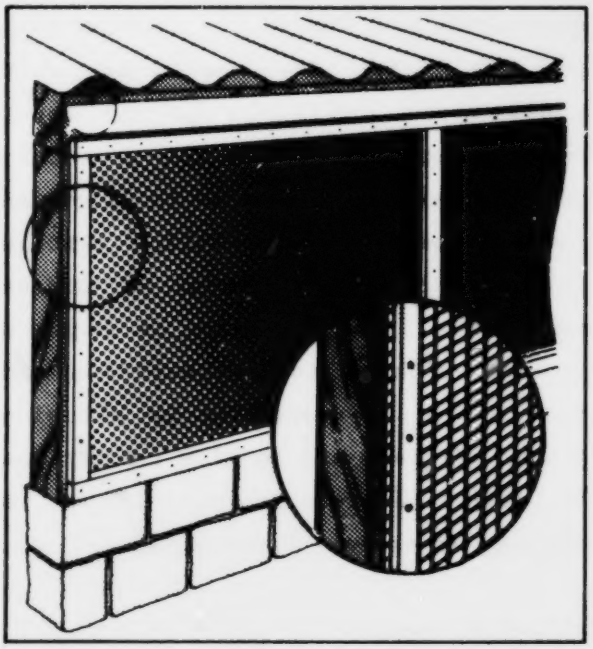


Figure 7.4 Tensar Cladding windbreak material.

Many well ventilated cattle barns have a series of 1200 mm x 2400 mm (48" x 96")

openings in the side wall spaced at 4800 mm (192") centres. The openings are protected in winter by solid panels hinged at the top or mounted on barn door tracks. See figure 7.1 for details.

To prevent drafty conditions inside the barn that may be caused by such large openings in windy weather, many producers are installing windbreak material. This may take the form of spaced boarding, consisting of 19 x 89 mm (1" x 4") wooden slats spaced with a 13 mm (1/2") gap. See figure 7.2. Another alternative is plastic mesh screen often referred to as "Turkey curtain". The trade product names are Lumite or Fabrene and are available at livestock equipment suppliers. For application instructions refer to Canada Plan Service leaflet number M-9351. See figure 7.3.

A new plastic product in B.C. is Tensar Cladding Windbreak. This is a strong, 50% porous mesh which allows the passage of light and fresh air but prevents drafts inside the livestock enclosure. It is available from Westcoast Polysystems Ltd., Mt. Lehman, B.C. (figure 7.4).

Ridge Vents

A ridge vent is critically important for naturally ventilated cold confinement barns. This allows the release of warm, moisture laden air which collects under the roof near the peak of the building. Even on still days when there is no wind, the stack effect still works to remove heat from the barn. Body heat from the animals warms the air, causing it to rise by convection to the top of the barn. As the warm air exits the ridge vent, fresh, cool air enters through the side wall inlets.

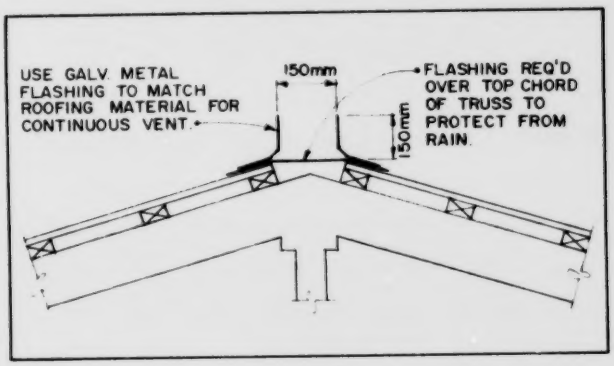

USE GALV. METAL FLASHING TO MATCH ROOFING MATERIAL FOR CONTINUOUS VENT.
150mm
150mm
FLASHING REQ'D OVER TOP CHORD OF TRUSS TO PROTECT FROM RAIN.

Figure 7.5 Open ridge vent design.

Several types of ridge openings have been used to accomplish this ventilation. One method that is common but ineffective is the use of attic-space vents; these "tin-hat" vents that are spaced about 6000 mm (20 ft.) apart along the ridge of a building. **Do not use this venting system.** There is insufficient outlet area provided and it is the wrong application for these vents.

The rule of thumb for ridge vents is the same as for side vents: **size of ridge opening is 50 mm per 3000 mm (2" per 10 ft.) of building width.** The opening

should be continuous for the entire length of the building except for the last 2400 mm to 3600 mm (8 to 12 ft.) at both ends of the barn.

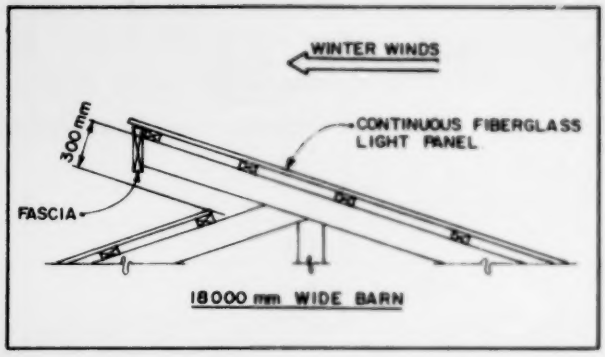


Figure 7.6 Overshot ridge vent design.

The most effective ridge vent is an open vent with vertical flashing to deflect wind upward. See figure 7.5.

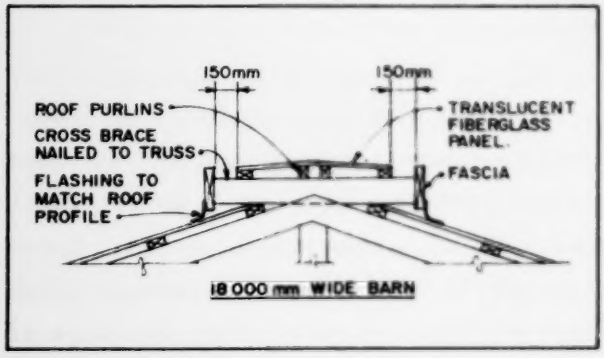


Figure 7.7 Protected ridge vent design.

There are situations where a ridge cap is desirable, particularly when it is raining heavily, the air is still and there is a barn full of fresh hay stored below the ridge. There are two designs which are common - the overshot and the protected ridge vent as shown in figures 7.6 and 7.7. The overshot ridge vent is not as expensive to construct but it offers protection from one direction only. Where light, wind driven snow is encountered, back-eddies created at the down wind side of the vent may allow snow to be blown into the barn.

Roof Insulation

A minimal amount of insulation (12 mm (½") Homasote, Dona-Cona, Ten-Test, Poly-styrene bead board, partical-board or plywood, or 38 mm (1½") foil-faced or vinyl-faced fibreglass blanket insulation) is often installed to correct ventilation problems in cold weather. Installing any form of roof insulation without first having provided the necessary wall and ridge openings, as discussed above, will do nothing to improve conditions within the barn.

Roof insulation does help to control condensation when conditions are severe and it is especially useful to reduce the radiation heat gain in the summer when the sheet-metal roofing reaches very high temperatures.

Pay careful attention to installation of soft insulation materials such as plastic foams or fibreglass. Birds can play havoc with such materials if any perch is provided within an inch or two of the insulation surface. Block all spaces

between roof straps and ensure all seams of fibreglass blanket material are tightly fitted.

Roof Slope

Since warm air rises, steeper sloped roofs provide for a higher velocity of upward air flow. Excessively steep roofs can lead to less fresh air in the animal zone since incoming air tends to rise too rapidly along the roof.

A good compromise is the 1:3 (4:12) roof slope which is most common for modern farm buildings. Roof slopes of less than 1:3 (4:12) are more prone to condensation and higher interior temperatures in summer.

Widely spaced structural members with deep purlins in excess of 150 mm (6 in) deep will usually lead to increased incidence of condensation. This is due to dead air pockets that tend to occur between the deep purlins.

TROUBLE SHOOTING NATURAL VENTILATION SYSTEMS IN COLD CONFINEMENT LIVESTOCK BARNS - from Bodman (1976)

The following are the 10 most frequently encountered problems:

1. Low roof slope (less than 1:3).

2. Little or no eave openings (25 mm per 3000 mm (1" per 10 ft.) of building width is recommended along both side walls).

3. Lack of eave opening on the closed side of a three-sided building as in the case of lean-tos.

4. Lack of or too small a ridge opening (50 mm per 3000 mm (2" per 10 ft.) of building width is recommended).

5. Ridge caps which are too wide or too low so as to prohibit continuous upward flow of air.

6. Ridge caps of proper configuration but the fascias or vertical flashings prohibit the continuous upward flow of air.

7. Lack of openable side wall panels for summer ventilation or panels which are too small (continuous row of 600 mm (24 in) panels are recommended, or equivalent).

8. Widely spaced trusses, rigid frames or arches with deep (over 375 mm deep) purlins.

9. Barn orientation so that prevailing winds are paralled to the ridge. Cross-flow ventilation is not possible in this case.

10. Building located in sheltered area. Cross-flow ventilation is not possible in this case.

Automatically Controlled Natural Ventilation (ACNV)

Although natural ventilation systems for cold environment barns are usually non-adjustable or manually adjustable, it is now possible to apply electro-mechanical controls to natural ventilation systems to achieve automatic control. For temperature control in intensive livestock housing, the amount of ventilation air is varied by opening and closing ventilation doors, flaps and vents. ACNV systems may offer advantages over conventional fan powered systems for the following reasons:

1. Systems that use variable speed fans are known to be susceptible to wind effects at low ventilation rates in winter.

2. Failure of electrical power can have disastrous consequences in hot weather since ventilation ceases unless some provision for natural ventilation is incorporated into the design of the building. Automatic start-up, stand-by generators are often installed to offset the potential loss of livestock in the event of power failure. If there is an electric power failure, the ACNV system stops in the mode in which it was last operating and ventilation continues but without control.

3. Fan systems consume large amounts of electric power at rates that continue to rise. Typical cost of fan operation in summer for a 500 hog finishing barn would be approximately $100 monthly. In comparison, while capital costs for the ACNV system will be competitive with a conventional fan ventilation system, the energy costs for ventilation will be negligible.

4. ACNV systems are quiet.

There are some potential disadvantages to the ACNV system:

1. In the summer, the system relies largely on the wind to provide cross ventilation to move sufficient volumes of air. If the air is still, the ridge vent will not exhaust air rapidly enough to control temperature.

2. It is difficult to predict how air will flow inside the barn which may cause messy pens in ACNV swine barns.

3. Some ACNV systems operate the side-wall flaps fully open or fully closed. The effect is similar to having a cycle timer or oversized fans in a conventional system. Temperature and relative humidity make sudden, large

fluctuations. Superior ACNV systems will open or close the ridge vent and side wall flaps in small stages to minimize the temperature fluctuations in the barn.

4. Rapid changes in environment with some ACNV systems leads to animal stress, disease, and decreasing growth rates.

5. There is a lack of experienced advice available on how to operate ACNV systems.

6. Winter performance of ACNV systems is usually poor. An additional mechanical ventilation system is recommended for the winter months.

Natural Ventilation Openings for Swine Barns

The total ridge opening area for gable roof swine barns should be 3200 to 3800 mm^2 (5 to 6 in^2) per finishing pig housed and 4500 mm^2 (7 in^2) per gestating sow. The opening should be at least 75 mm (3 in) wide to prevent frost build-up in extremely cold weathers. Do not add ridge caps that reduce air-flow and trap snow. Flash the exposed roof purlins and truss upper-chords. If birds are a problem install 20 mm x 20 mm (3/4" x 3/4") galvanized hardware cloth over the ridge openings. It may be necessary to knock the ice off screened openings in extremely cold weather.

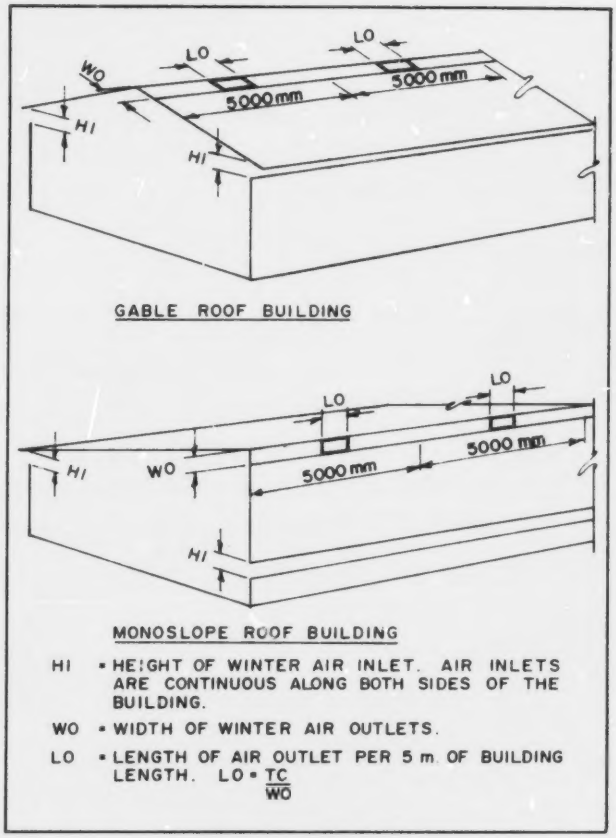


Figure 7.8 Natural ventilation openings.

The eave openings should be continuous down both sides of gable-roof barns. The total eave opening should be at least as large as the total ridge opening. Install 20 mm x 20 mm (3/4" x 3/4") galvanized hardware cloth to keep birds out. The eave openings can be provided by the partially opened side-wall flaps or by the open space above the top plate of the stud wall. Baffles should be provided so airflow can be reduced in severe storms.

Table 7.1 Natural Ventilation Openings For Swine*

7.1(a) Slot widths for cold and mild weather ventilation

Install a continuous **winter air inlet** along each side of the building. Make each inlet the height (HI) in this table.

Make **winter air outlets** at least 80 mm (3") wide to prevent them from freezing shut. It is usually necessary to install several shorter vents that are 80 mm or wider, instead of a continuous narrow opening. Provide air outlets at least 5000 mm (16') o.c. To determine the length of the outlet opening required per 5000 mm of building length (LO), divide the table constant (TC) by the outlet opening (WO = 80 mm or wider).

Building Width mm (ft)	Winter air inlet height (HI)		Winter air outlet constant (TC)	
	Finishing mm (in)	Gestation mm (in)	Finishing -	Gestation -
3000-4600 (10-15)	19 (0.75)	13 (0.50)	12	8
4900-6100 (16-20)	25 (1.00)	19 (0.75)	16	12
6400-7600 (21-25)	32 (1.25)	25 (1.00)	20	16
7900-9100 (26-30)	38 (1.50)	32 (1.25)	24	20
9500-10700 (31-35)	44 (1.75)	38 (1.50)	28	24
11000-12200 (36-40)	51 (2.00)	38 (1.50)	32	24

7.1(b) Side wall openings for warm weather ventilation

Heights given are for one opening the entire building length. Provide an opening in **each** side wall.

Building Width mm (ft)	Gable roof	Monoslope roof	
	each sidewall mm (in)	Backwall mm (in)	Frontwall mm (in)
3000-4600 (10-15)	610 (24)	310 (12)	910 (36)
4900-6100 (16-20)	760 (30)	410 (16)	1220 (48)
6400-7600 (21-25)	910 (36)	460 (18)	1520 (60)
7900-9100 (26-30)	1070 (42)	610 (24)	1680 (66)
9500-10700 (31-35)	1220 (48)	-	-
11000-12200 (36-40)	1520 (60)	-	-

* From Midwest Plan Service Swine Handbook Housing and Equipment, 1983, MWPS-9. Iowa State University, Ames Iowa.

The openings should never be completely closed. Several types of vent flaps work well including top and bottom hung doors and pivoted doors. The flaps should open to a full horizontal position to provide the maximum possible summer air-flow.

Table 7.1 and figure 7.8 are useful for designing ventilation opening sizes. ACNV systems should conform to these guidelines to ensure adequate ventilation potential. If alternate configurations are used, total opening area should be equivalent to those recommended in table 7.1.

CHAPTER 8 POLY-TUBE VENTILATION

Poly-tube (or fan-jet) ventilation systems are proving to be very effective in remodelled buildings where the installation of adjustable air inlets may be difficult or impractical. Poly-tubes can be used where additional air circulation is required, especially under the low air flow conditions of winter operation. Poly-tubes are also used very successfully to provide fresh air in cold-environment calf rooms or cattle barns.

An inlet fan pushes air through a perforated plastic duct (poly-tube) which distributes the air down the full length of the room to be ventilated. The inlet fan is usually spaced about 1 fan diameter away from the end wall where a set of inlets shutters may be located. When air conditions inside the barn are near the thermostat set-point, the inlet shutters are closed and air within the room is recirculated. If cooling is required, the motorized inlet shutters open and, simultaneously, the exhaust fan comes on. The inlet shutters must always be cracked open so that at least a minimum quantity of fresh air is pulled into the room.

Unit heaters can be located near the inlet fan so that heated air can be pushed down the tube.

Another common configuration is to use the **poly-tube as an inlet only** in conjunction with exhaust fans. For relatively short tubes, no inlet fan is required. This set-up is particularly useful for distributing heated air from a pre-heated hallway. No recirculation is possible with this configuration.

For **cold-environment barns**, poly-tubes can supply fresh air to cattle pens that have inadequate natural ventilation. Air is distributed through the barn through one or more poly-tubes fitted to inlet fans mounted directly on the wall. Air exhausts though openings in the building, hence no exhaust fans are normally required.

Guidelines for design and installation of poly-tubes

1. Size poly-tube systems to satisfy the minimum or winter air-flow requirements of the livestock or poultry housed. (To satisfy additional air-flow add larger poly-tubes as required, use conventional negative pressure

ventilation systems, or rely on natural ventilation. For most situations in temperate climates, the winter period is the critical period when poor ventilation is experienced.)

2. Table 8.1 shows the recommended tube and fan combinations for poly-tube systems.

3. **Number of holes.** Too few holes prevent the fan from delivering its full capacity and too many holes prevent the duct from inflating. The total area of holes is critical for any given tube regardless of how long the tube may be. The important factor is the aperature ratio - the ratio of the total area of holes divided by the cross-sectional area of the tube. The recommended aperature ratio is 1.5. Table 8.2 may be used for poly-tube design.

   Common practice in industry is to use an aperature ratio of 0.5 to 1.0. If a fan is attached to a poly-tube with an aperture ratio of 0.6 the fan will operate at about 60% of its potential maximum capacity and the static pressure in the duct will be about 6 mm (1/4 in.) water guage. By increasing the aperture ratio to 1.5 the capacity of the fan increases to near 100% of its rated capacity and the mean static pressure in the poly-tube will be in the order of 3 mm (1/8 in.) water guage. Some designers specify hole size based on an average jet velocity from each hole of about 4 m/s (800 fpm). This neglects the vena contracta effect of the air jet. In reality, the effective hole size will be less than half of that calculated.

   The poly-tube will remain stable and inflated, provided it is straight and suspended from its top edge, even if the fan speed is reduced to 20% of its maximum rated speed. This is an important consideration for use of variable speed fans with poly-tubes.

4. **Size of holes.** 50 mm (2 in.) diameter holes is a practical size that will give "throw" to the individual jets of air. To avoid drafts on young animals use smaller holes. 25 mm (1 in.) holes are recommended for 300 mm (12 in.) diameter poly-tube installations in calf-barns and swine nurseries.

5. **Positioning the holes.** The discharge of air from each hole is not uniform along the length of the duct. For short ducts, more air is discharged from holes furthest away from the fan. For longer ducts, friction effects come into play and air discharge is nearly uniform along the

duct or, very commonly, the discharge can be greatest near the fan. Until better experimental data is available, the best recommendation is to space the holes uniformly along the length of duct.

Holes can be placed in any pattern around the duct in order to discharge the air in the desired direction. It is safest to direct the air sideways in one or both directions. Direct air toward dunging areas in loosely penned animal housing. Direct air to the front of the animals where they are tethered in one position.

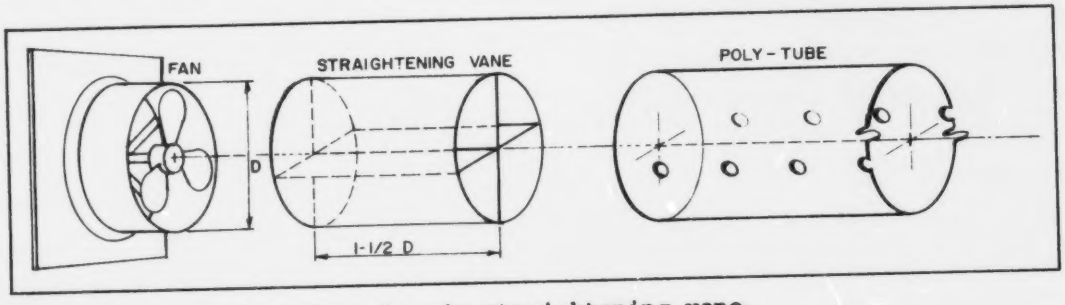


Figure 8.1 Simple poly-tube air straightening vane.

6. **Directional effects of air discharging from the holes.** Air is discharged in a downstream direction and there is little that can be done to counteract this effect. Suggestions have been made to cut D-shaped holes rather than round holes so that the remaining flap of plastic, attached to the duct downstream of the hole, acts as a deflector. The effort to do this does not seem to be warranted.

   A second directional effect is tangential discharge from holes, particularly from those holes near the fan. This is due to the swirl of the air induced by the rotating fan blades. The effect is easily overcome by fitting an air-straightener between the fan and the duct inlet.
   The length should be 1.5 times the fan diameter and the best design will have radial fins - at least 4 and preferably 8. The fins need not extend to the centre. A simple cross made up of two interlocking panels of sheet metal is better than nothing. (See figure 8.1.)

7. **Suspension of the poly-tube.** Most commonly, plastic straps are used to suspend the duct but this is not satisfactory if the inflation of the tube is to be reduced by reducing the speed of the fan or shutting off power to the fan. When the inflation is increased again with an increase

in fan speed, the poly-tube is left looking like linked sausages. It is recommended that the poly-tube be suspended from the top edge using a series of 75 mm (3 in.) "bull-dog" paper clips spaced at 1200 mm (4 ft.) (figure 8.2). It is necessary to insert a small dowel for the bull-dog clip to clamp onto, inside the poly-tube. A superior method is to use vinyl-coated clothesline wire that is passed inside the entire length of the duct. The wire is suspended from hooks at 3000-6000 mm (10 to 20 ft) centres by making a small cut in the top edge of the duct.

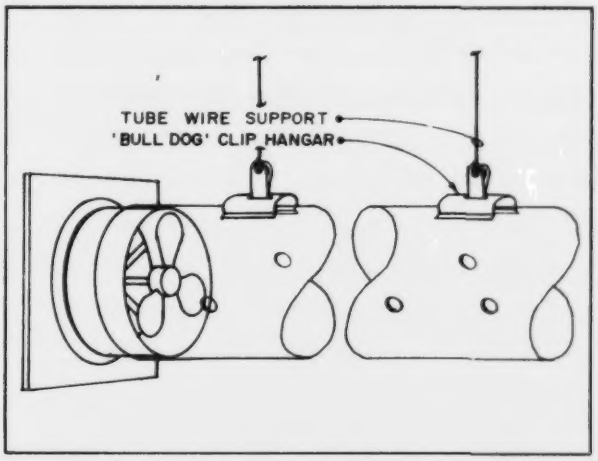


Figure 8.2 Suspension of the poly-tube using "Bull-dog" clips. Clip the poly-tube using a 6 mm (1/4 inch) wooden dowel inside the poly- tube so that the clip does not slip off the plastic.

TABLE 8.1 COMMON POLY-TUBE AND FAN COMBINATIONS

Duct Diameter	Fan Motor Horse Power	Air Flow	Maximum Length
SI			
300 mm	125 - 190 W	500 - 570 L/s	30 m
450 mm	190 - 250 W	1400 - 1500 L/s	60 m
600 mm	250 - 375 W	2350 - 2550 L/s	110 m
750 mm	375 - 560 W	3775 - 4000 L/s	120 m
IMPERIAL			
12 in.	1/6 - 1/4 HP	1000 - 1200 cfm	100 ft.
18 in.	1/4 - 1/3 HP	3000 - 3200 cfm	200 ft.
24 in.	1/3 - 1/2 HP	5000 - 5400 cfm	360 ft.
30 in.	1/2 - 3/4 HP	8000 - 8500 cfm	400 ft.

TABLE 8.2 POLY-TUBE DESIGN CHART

Duct Diameter	Duct Area	1.5 x Duct Area								
			Diameter of Holes (mm)							
SI			20	25	30	40	50	65	75	100
			Number of Holes							
300 mm	0.0707 m^2	0.1060 m^2	338	216	150	84	54	--	--	--
450 mm	0.1590 m^2	0.2386 m^2	--	486	338	190	122	72	54	--
600 mm	0.2827 m^2	0.4241 m^2	--	--	600	338	216	128	96	54
750 mm	0.4418 m^2	0.6627 m^2	--	--	--	527	338	200	150	84
			Diameter of Holes (inches)							
IMPERIAL			3/4	1	1 1/4	1 1/2	2	2 1/2	3	4
			Number of Holes							
12 inch	113 in^2	170 in^2	385	216	138	96	54	--	--	--
18 inch	254 in^2	381 in^2	--	485	310	216	121	78	--	--
24 inch	452 in^2	678 in^2	--	--	552	384	216	138	96	54
30 inch	706 in^2	1060 in^2	--	--	--	600	337	216	150	84

CHAPTER 9 HEAT EXCHANGERS

Heat exchangers are capable of recovering heat which would otherwise be exhausted out of the barn in the form of warm air. Warm exhaust air is passed through the device at the same time as cool outside air is brought back in; the warm air serves to partially preheat the air entering the barn. Energy is thereby saved.

The questions frequently asked of heat exchangers are: "Do they work?" and "Do they make economic sense?" The answer to both questions is "yes, but not in all cases." The colder the weather, the better are the prospects of a heat exchanger being economical.

The purpose of this chapter is two-fold. First, the necessary technical information for evaluation and comparison of heat exchangers is presented and second, a technique for evaluating the economics of heat exchangers is defined.

HEAT EXCHANGER THEORY

General

When outside temperature drops below a certain point called **the heat deficit temperature**, the heat lost by the building to the outside environment is greater than the heat produced by the animal livestock. Supplemental heat is thus required if the barn temperature is to remain constant.

Part of the building heat losses to the environment are contained in the warm ventilation exhaust air. Under normal circumstances this heat is unrecoverable.

By using a heat exchanger, a percentage of ventilation heat losses can be recovered. Since heat losses are lower, less supplemental heating is required. There are therefore two benefits to a heat exchanger: the capital cost of supplemental heating equipment will be reduced or eliminated, and the running cost of heat energy will be reduced.

The running costs of a heat exchanger are the electrical energy required to run its fans and the labour required to maintain it. Frequent cleaning is required for heat exchangers as their performance is severely affected by dust and dirt build-up.

Types of Heat Exchanger

At the present time only one basic kind of heat exchanger is used in livestock structures. This is the air-to-air heat exchanger. Other design concepts such as regenerative rock bed, and earth-tube heat exchangers are not adequately refined for general use.

Air-to-air heat exchangers may be categorized as either **counter-flow**, **cross-flow**, or **parallel-flow**. Heat exchange in any of these cases is through plastic or metal plates (Figure 9.1). Theoretically, the counter-flow type of heat exchanger has the highest effectiveness, the cross-flow second, and the parallel flow third (Figure 9.2).

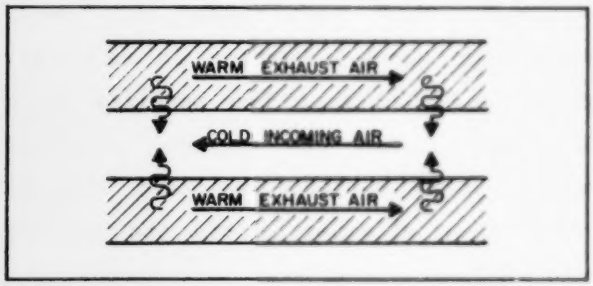


Figure 9.1 Heat transfer across the exchange plates in a heat exchanger.

Metal plate heat exchangers are generally not practical for use in agricultural applications. Corrosion of the plates due to high exhaust air acidity and severe frosting problems have led to the nearly exclusive use of plastic in the core construction of agricultural heat exchangers.

Condensation occurs continuously in the heat exchanger. Warm exhaust air having high moisture content gives up its heat to the incoming air and condensation forms on the exchange plate surface. Since the heat exchanger is meant for use in cold weather conditions, frosting and freeze-up of condensed waters can be a problem.

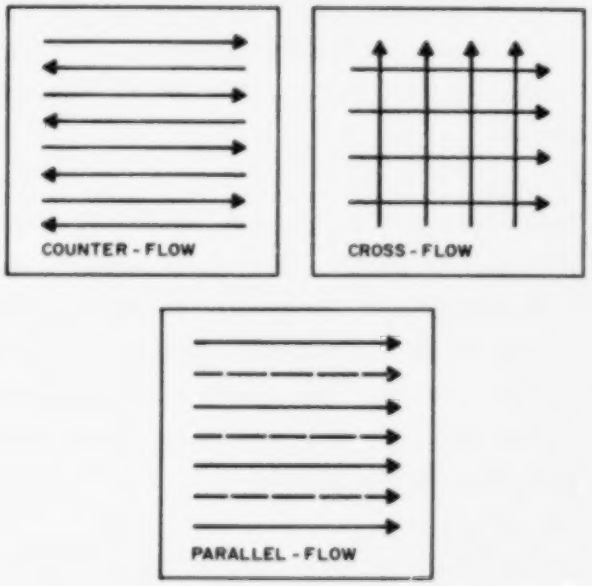


Figure 9.2 Types of air-to-air heat exchangers.

Different systems use various methods of defrosting the exchange plates; reversing the air flow periodically is the most common solution. Depending on how the condensed water is removed, ice build-up around the water discharge outlet can also cause difficulties.

Dust and dirt build-up in the heat

exchanger seriously reduces the effectiveness of the unit. Heat transfer across the exchange plates diminishes significantly with dust build-up. Cleaning of the heat exchanger is required frequently. Designers usually provide for some relatively easy means of cleaning, but some systems are better than others. A pressure washer is usually necessary to carry out the cleaning operation.

Definitions

Various performance parameters may be used to evaluate heat exchangers. Definitions are provided below.

Effectiveness is commonly cited by manufacturers. It is useful for design and selection of heat exchangers, but is misleading as an indicator of heat exchanger performance. It identifies only the effectiveness of raising incoming air temperature; it does not indicate the amount of heat actually exchanged since it does not include air flow rate in its definition.

(Eq. 9.1)

$$\text{Effectiveness} = \frac{t_e - t_o}{t_i - t_o}$$

t_e = temperature of incoming air after passing through the exchanger, °C or °F.

t_o = outside air temperature, °C or °F.

t_i = inside room air temperature, °C or °F.

Average heat recovery is the average rate at which sensible heat is recovered from the exhaust air at a given standard condition. The average heat recovery indicates how much heat can be recovered by a heat exchanger under real operating conditions, including the defrost cycle when no heat is recovered. It is not a comparision parameter for different heat exchangers unless they have identical exhaust capacities. Average heat recovery is measured in units of kW or Btu/hr.

Exhaust heat loss is the total heat lost through the heat exchanger due to heat contained in the exhaust air flow and may be determined from equation 2.15 found in chapter 2.

(Eq. 2.15)

$$q = \frac{Q}{v_s}(h_i - h_o)$$

q = exhaust heat loss, kW or Btu/hr.

Q = air flow volume, m^3/s or ft^3/hr.

v_s = specific volume, m^3/kg da or ft^3/lb da.

$h_i - h_o$ = enthalpy difference between inside and outside air, kJ/kg da or Btu/lb da.

Heat recovery ratio is the best means of comparison between different heat exchangers. It is the ratio of average heat recovery to average heat loss expressed as a percent. The higher the heat recovery ratio, the better is the performance of the heat exchanger.

(Eq. 9.2)

$$Hr = \frac{\text{Average heat recovery}}{\text{Exhaust heat loss}} \times 100$$

Hr = heat recovery ration, %

Heat exchanger airflow rates are the **exhaust** and **supply** airflow rates for the heat exchanger. They are important for sizing of the exchanger.

Leakage is the amount of air that leaks from the exhaust to the supply airstream during operation. It is usually expressed as a percent.

Power requirements for the operation of the unit are usually minimal but should be taken into consideration.

AIR DISTRIBUTION

The first point to remember with heat exchangers is that they must operate in concert with the normal ventilation system. Heat exchangers will not handle the total ventilation requirements of the barn. **Other air inlets and exhaust fans must be provided.**

The other point to remember is that heat exchangers introduce supply air into the barn at a point source. Mixing and distribution of this air is still necessary. **A heat exchanger should not be installed to improve poor environmental conditions in a barn.** It is usually more cost effective to make improvements to the existing ventilation system first. A new heat exchanger must operate compatibly with the existing ventilation system and until the necessary improvements are made it will probably not do this.

Drafts can be a real problem with heat exchangers and supplementary air diffusers or circulation systems should be a part of the system design. Methods of solving the air circulation problem include:

- Vent the heat exchanger into the preheat room or hallway if one exists.
- Duct the air from the heat exchanger to a duct above a centre ceiling air inlet and use inlet as normal.
- Use the circulation tube system if one exists. A poly-tube ventilation system premixes room air with cold outside air to prevent condensation on the tube surface. Vent the heat exchanger exhaust to the premix manifold.
- Provide recirculating fans.

SIZING THE HEAT EXCHANGER

There are three sizing methods used for heat exchangers.

1. **Selection on the basis of the minimum continuous ventilation rate.**

   In this case the heat exchanger will be returning some of the ventilation heat losses to the barn as supplemental heat. This may not be enough heat to maintain heat balance under all outside conditions. Other supplemental heating should still be provided. Where a heat exchanger is being installed in an existing facility this is probably the best option.

2. **Selection on the basis of two-thirds of the maximum instantaneous heating demand.**

   In a new facility this method will reduce the capital cost of the space heating system and help offset the cost of the heat exchanger(s). The other one-third of the instantaneous heating demand must still be provided for by the supplemental heating system. By this sizing scheme the heat exchanger will provide about 90% of the seasonal heating requirement since heating demand is rarely at the instantaneous maximum.

3. **Selection on the basis of the full supplemental heating requirement to meet a given inside temperature and relative humidity.**

   This method permits the complete replacement of the supplemental heating system with heat exchangers. It also provides the capacity to limit relative humidity to levels less than the usual 75% to 80% which are the basis of the minimum ventilation rates. **Complete heat balance information on the barn is required for this determination.**

ECONOMICS OF THE HEAT EXCHANGER

In general, only very cold climates will allow for a heat exchanger to pay for itself on the basis of energy recovery. Heat exchangers in the South Coastal Region of B.C., for example, really don't merit much thought. In Fort St. John, however, heat exchangers are often a good investment.

Economic evaluations may be simple or complex but they all attempt to address three basic questions: the capital cost difference between systems, the operating cost difference, and the additional benefits which may arise out of the use of one system or the other.

The capital cost difference is relatively easy to determine once designs for a system with and without heat exchangers have been completed.

In order to determine operating cost

savings some information must be collected. This information includes **weather data** for the region of concern, **supplemental heating requirements for the barn**, the **cost of energy** and the **performance characteristics of the heat exchanger**.

Additional benefits include improved animal health, extended building life due to lower RH levels, fewer cold drafts since inlet air is no longer freezing, and elimination of the fogging problem.

Any economic evaluation is based on assumptions. The analysis is only as valid as the assumptions are reasonable. Calculations of the economics of heat exchangers are extremely sensitive to the following parameters.

1. The heat deficit temperature (the outside temperature at which building and ventilation heat losses are greater than animal heat production);

2. The actual, average, or worst case weather data;

3. The heat exchanger effectiveness.

Economic evaluations of heat exchangers can wrongly indicate a cost benefit when other more cost effective measures are not taken. For example a poorly insulated barn can easily indicate that a heat exchanger is a good idea when it would be better to properly insulate the building. One should address the obvious energy saving measures of proper insulation, energy efficient motors, and heating/ventilation interlock before arriving at the conclusion that heat exchangers are the answer to the farmer's energy conservation problem.

Economic Calculations

Figure 9.3 shows degree-day curves for several areas of British Columbia. These curves provide the necessary weather data for a simplified economic evaluation.

The first step in an economic evaluation of heat exchangers is to determine the heat deficit temperature of the barn. Heat deficit temperature may be determined by rearranging equation 2.16 found in chapter 2 to the form of equation 9.3.

(Eq. 9.3)

$$HDT = t_i - \frac{(v_s/c_p)q_s}{Q_{min} + (v_s/c_p)(A/R)}$$

HDT = Heat deficit temperature, °C or °F

t_i = Inside barn temperature, °C or °F

Q_{min} = Minimum winter ventilation rate, m^3/s or ft^3/hr (not cfm)

q_s = Sensible heat production of livestock, W or Btu/hr

v_s = Specific volume of air, m^3/kg or ft^3/lb

c_p = Specific heat of air, approximately 1000 J/kg-°C or 0.24 Btu/lb-°F

A/R = Building heat loss factor, W/°C or Btu/hr-°F

In order to use equation 9.3 it is necessary to find the values for all the parameters. These determinations are discussed in other sections of this handbook with the exception of the building heat loss factor. A/R may be computed using values found in the Agriculture Canada publication **Insulation in Farm Building**, publication 1601/E, 1983.

(Eq. 9.4)

$$A/R = A_T \sum_{i}^{n} \frac{1}{R_i}$$

A_T = Total area of walls, ceiling, foundation perimeter and floor.

$\sum_{i}^{n} \frac{1}{R_i}$ = The net sum of the inverse of all R values for the walls, ceiling, foundation perimeter and floor.

The second step is to use figure 9.3 to determine the number of degree-days below the heat deficit temperature. Read HDT on the temperature axis of figure 9.3. Then read the celsius degree-days below HDT at the point of intersection with the geographic location of choice.

Third, estimate the seasonal supplemental heating requirements with equation 9.5.

(Eq. 9.5)

SI

Energy consumption (J) = 86400 $c_p D Q_{min}/v_s$

Imperial

Energy consumption (Btu) = 1440 $c_p D Q_{min}/v_s$

D = Degree-days below heat deficit temperature, °C-days or °F-days

Q_{min} = Minimum winter ventilation rate, m^3/s or cfm

Fourth, find the effectiveness of the exchanger for the expected inside/outside temperature differential. This is usually taken as the difference between inside temperature and the heat deficit temperature.

Fifth, determine the potential energy cost saving with equation 9.6.

(Eq. 9.6)

SI

Energy savings =
(Effect.)(Energy Consumption)(Energy Cost)

Imperial

Energy savings =
(Effect.)(Energy Consumption)(Energy Cost)

Energy savings = $/yr
Effectiveness = express as a decimal
Energy cost = $/J or $/Btu
Energy consumption = seasonal energy requirements, J or Btu per year

Sixth, determine the values of the other important variables: the running costs of the heat exchanger (electricity cost plus maintenance costs), the capital cost premium of the heat exchanger - cost of exchanger less the cost of heaters which are eliminated (applies only to new buildings), and an estimate of the intangible benefits such as improved animal health.

Finally, determine the net savings and calculate the simplified payback period:

Net annual saving = Energy savings - running costs + other benefits

$$\text{Payback period (yrs)} = \frac{\text{Capital cost premium}}{\text{Net annual saving}}$$

If the payback period is less than 5 or 6 years, the heat exchanger is probably cost effective. Before the conclusion is reached, however, problems with the evaluation presented above should be considered.

Attaching a value to the "intangible benefits" is impossible. It is better to err on the safe side of an economic evaluation and ignore such benefits.

Estimating running costs of a heat exchanger is largely guesswork. Calculating the energy cost of the heat exchanger fan motors is simple enough but the valuation of maintenance costs is not. Some farmers don't attach a value to their time and would gladly spend a few hours a week doing maintenance work to save a few dollars on hydro bills. Another factor is the pressure washer which is necessary for maintenance - if the farmer doesn't already have one, some portion of its cost should be applied.

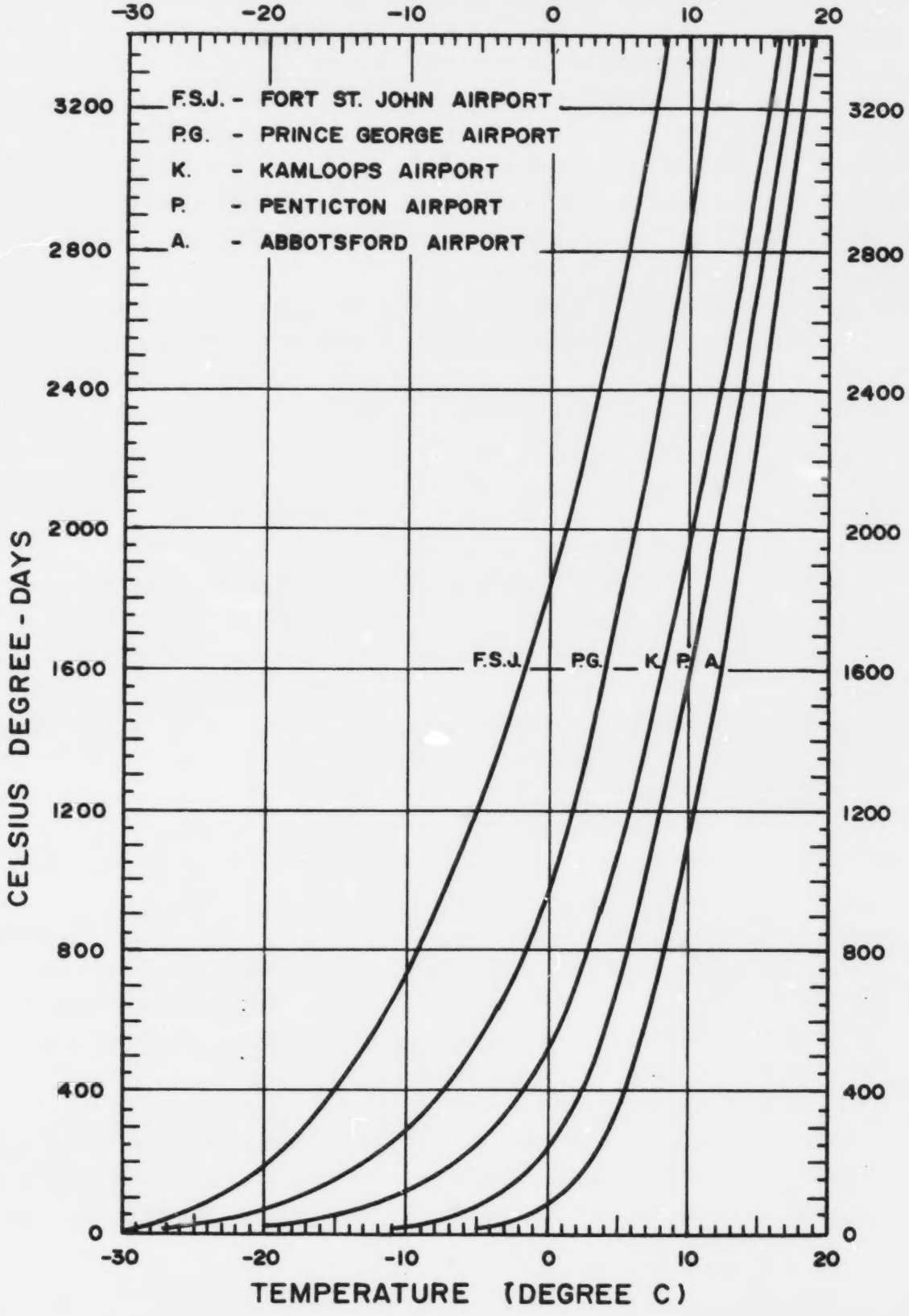
Equation 9.5 is based on the minimum ventilation rate; a larger portion of heat may be recovered if the exchanger is oversized (sizing method 2 or 3). Another assumption in equation 9.5 is that the Effectiveness rating of the exchanger is constant - dirty exchangers will have a much lower Effectiveness.

Finally, one should consider some of the inadequacies of the Simple Pay-back evaluation. More sophisticated calculations include such factors as interest rate, present value of future cash flow, taxes applied to profits, and depreciated value of the heat exchanger itself.

In view of the large number of variables affecting the economics of heat exchangers, it is wise to make conservative estimates

of the energy benefits. When the calculations still show a clear cost saving, the use of heat exchangers will be beneficial.

Figure 9.3 AVERAGE ANNUAL HEATING DEGREE-DAYS After B. van Dalfsen, 1982



CHAPTER 10 COOLING

High summer temperatures can cause heat loading on agricultural buildings which cannot be adequately countered by increased ventilation. Heat stress on animals and poultry occurs under such conditions and leads to diminished production and higher mortality rates. Evaporative cooling equipment can be installed to reduce the heat stress caused by high summer temperatures.

EVAPORATIVE COOLING PRINCIPLES

Evaporative cooling is a process in which hot, unsaturated air is brought in contact with free water. Because the free water surface has a higher vapour pressure than that of the unsaturated air, water is transfered to the air. The change of state from liquid to vapour requires heat of vaporization which is provided by sensible heat contained in the unsaturated air. The net transfer of energy is zero, but the dry-bulb temperature of the air drops as relative humidity increases. This cooler air provides relief to heat stressed animals and birds.

A psychrometric chart is used to follow the evaporative cooling process. Figure 10.1 shows the saturation of air as it proceeds along a constant wet-bulb temperature line. Point A defines the state of the outside air before it passes through the evaporative cooler. Point C describes the same air if it were fully saturated (i.e., RH = 100%). Since no evaporative cooler is perfectly efficient, the air leaving the cooler only reaches the condition at point B.

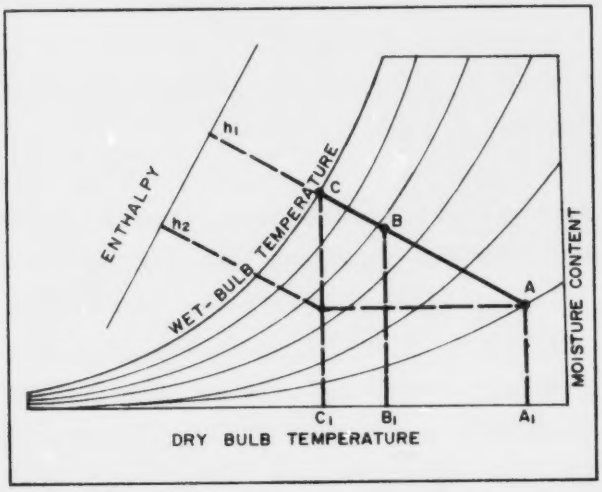


Figure 10.1 Evaporative cooling process.

Evaporative cooler efficiency is a term which is used to describe the saturation efficiency of a cooler.

$$E_c = \left[\frac{A_1 - B_1}{A_1 - C_1}\right] \times 100 \qquad \text{(Eq. 10.1)}$$

A, B, C = temperatures shown on figure 10.1

E_c = evaporative cooler efficiency, %

The term E_c has limited utility since it does not say what cooling capacity the unit has. Better measures of cooler performance include the quantity of air being passed through the cooler.

Wet-bulb depression is the potential temperature drop for a given condition (A_1 - C_1 on dry bulb temperature line). **Potential temperature drop** is frequently referred to when a specific geographic location is being considered for the use of evaporative coolers. If humidity in an area is generally high, then potential temperature drop will be relatively small; the higher the average relative humidity, the less effective is the evaporative cooler.

Evaporative Cooler Performance (ECP) measures the equivalent refrigeration of a cooler in heat units; in other words, the amount of heat energy involved in the adiabatic exchange. The formula for ECP is the same as equation 2.15 and the process is shown in dotted lines on figure 10.1.

(Eq. 10.2)

$$ECP = \frac{Q}{v_s}(h_2 - h_1)$$

ECP = heat added, kW or Btu/min

Q = airflow volume, m^3/s or ft^3/min

v_s = specific volume, m^3/kg da or ft^3/lb da

h = enthalpy of air, kJ/kg da or Btu/lb da

Since ECP will vary for any particular location (due to variation in relative humidity levels) a better measure is **Unit ECP**.

(Eq. 10.3)

Unit ECP = ECP/wet-bulb depression

Unit ECP allows ready comparison of evaporative cooler performance regardless of local conditions.

When determining average values for wet-bulb depression in a given locality, it is wise to keep in mind that humidity drops as temperature rises and is usually lowest during the hottest part of the day. As a consequence evaporative coolers generally provide their best performance when it is needed most - during high afternoon temperatures. In humid regions the justification for evaporative cooling is often in reducing brief afternoon periods of extreme heat stress (and hence reducing mortalities). While average potential temperature drop in humid areas will not be as large as in arid areas, evaporative cooling may still be effective.

TYPES OF EVAPORATIVE COOLING SYSTEMS

Evaporative Cooling Pads

Pad type coolers are the most popular method of evaporative cooling. Air is passed through a wet porous material, or pad, which presents a large wetted surface area to the air flow. Vapour transfer on such pads is usually good to excellent when the pads are new and clean, but dirt and mineral build-up seriously reduces cooler efficiency. **Periodic cleaning and replacement of the pad material is necessary for effective operation.**

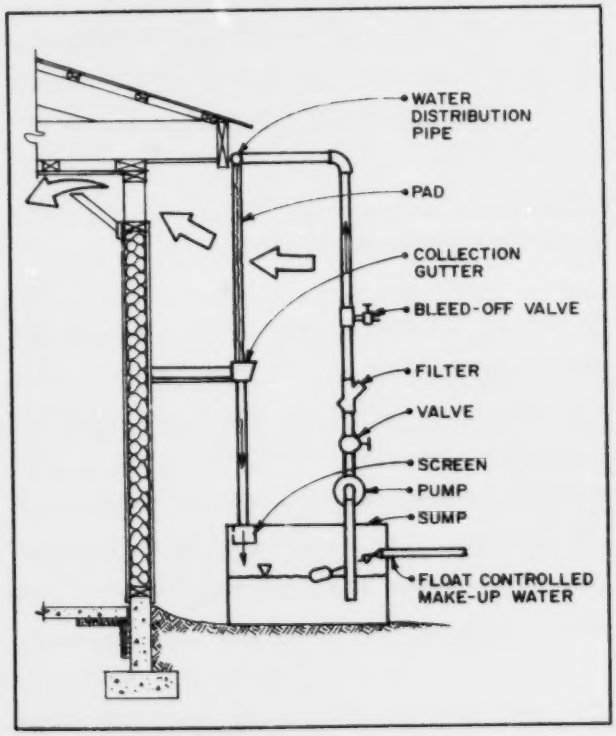


Figure 10.2 Vertical pad evaporative cooler.

Pad material had traditionally been **Aspen wood excelsior** which is effective for a single season provided it is cleaned mid-season to remove filtered dust. Excelsior pads require replacement once a year. Many other pad materials have been tried and are composed variously of wood, metal, mineral, glass, plastic and cement. Good pad design attempts to provide a maximum porous wetted surface, resistance to decay, and high capillarity (wick-like quality). Less porous materials such as metal have excellent resistance to decay but poorer saturation efficiency.

Newer pad materials are available made of a composite containing cellulose, plastic, anti-rot chemicals, and wetting agents. Pads such as the ACME Kool-cel are designed to last for as much as five years before replacement is required. Saturation efficiencies are reportedly higher for cellulose pads than the traditional excelsior, which reduces the required pad area, but cost per unit area is also higher. Limited data is available on actual performance of these pads after the first year.

Pad sizing and design

Pad orientation, pad area and thickness, air velocity, and the pad wetting system are all important design parameters for this type of evaporative cooling system. For vertical pad orientations figure 10.2

is a good design. Other wall mounted designs omit the air plenum and are installed directly in wall openings. These are less expensive to install but have the disadvantage of being part of the wall year round. Fall maintenance of the system shown in figure 10.2 simply involves removal and storage of the pads for the winter; no wall openings need be closed off or insulated for the winter.

Several other pad locations have proven to be ineffective and should not be used. **Roof-mounted**, and **tower and plenum systems** are extremely difficult to maintain and are **not recommended**.

Horizontal pad orientations with single or multi-level arrangement of pads have been used to some advantage. Water is sprayed onto the horizontal pad and serves to both wet and wash the pad surface. Dust build-up is decreased with this system. An additional advantage of multi-level pad arrangements is that the required pad area fits in a restricted width and pads can be mounted for easy access and removal.

To determine total required pad area the following formulas from Midwest Plan Service **Structures and Environment Handbook**, 1983, may be used.

Excelsior pads

(Eq. 10.4)

SI

$$A = \frac{\text{summer ventilation rate (L/s)}}{760}$$

Imperial

$$A = \frac{\text{summer ventilation rate (cfm)}}{150}$$

Cellulose pads

(Eq. 10.5)

SI

$$A = \frac{\text{summer ventilation rate (L/s)}}{1270}$$

Imperial

$$A = \frac{\text{summer ventilation rate (cfm)}}{250}$$

A = pad area, m^2 or ft^2

Pads are generally available in 100 mm (4 in) and 150 mm (6 in) thicknesses. The advantage of increased thickness is that air contact time with the wetted pad surface increases and improves saturation efficiency. It is unknown what precise pad thicknesses are optimum, however the 100 mm pad is most popular.

The pad wetting and water recirculation system is extremely important. Mineral build-up in the pad will rapidly diminish

efficiency unless excess salts are leached out of the pad by passing more water through the pad than is evaporated. If evaporation exceeds the wetting rate then mineral salts will build-up and block the pores in the pad. Water recirculation rate must be calculated to be in excess of the highest possible evaporation rate.

To prevent mineral build-up in the recirculating water, some bleed-off of sump water is required. Bleed-off can be continuous, or the sump can be emptied periodically and refilled with clean water. **Plugging of bleed-off valves is a problem and such valves require regular cleaning.** The bleed off rate is usually taken to be 8 L/h for each m^3/s of air flow (1 gal/hr for each 1000 cfm).

Table 10.1 shows recommended rates of water circulation and minimum sump capacity for typical conditions. Sump capacity can safely be increased but water rate should not be increased above 150% of the tabulated values. Excessive application of water causes the pad to be covered in a "sheet" of water and reduces surface to air contact area.

Installation and maintenance points

- Locate pads on the side wall opposite exhaust fans; preferable location is on the summer windward side of the barn with the pads centered on the wall.
- Sump construction should be of a rust free material: concrete, aluminium or a lined tank should be used. Steel tanks are too rusty.
- Use a shut-off float valve to control sump water level.
- Screen and filter the recirculating water.
- Use a bleed-off valve at the recommended rate or empty the sump completely once a month.
- Wet pads can weigh about 13.5 kg/m^2 (2.75 lb/ft^2); make sure the pad is adequately supported and reinforced.
- Clean the accumulated sediments from the sump once a year.
- Wire the thermostat control to stop pump operation 10 minutes before the fans cut out - this dries out the pads and prevents algae.
- A copper sulfate solution helps control algae; light tight sump construction also prevents algae build-up.

Table 10.1 Recommended Water Flow and Sump Capacity*

Pad Type	Minimum Water Rate per Linear Length of pad		Minimum Sump Capacity per unit pad area	
	L/h-m	(gpm/ft)	L/m^2	(gal/ft^2)
Aspen Excelsior 50-100 mm (2-4 in)	240	(0.3)	20	(0.5)
Corrugated cellulose 50-100 mm (2-4 in)	360	(0.5)	30	(0.8)
Corrugated cellulose 150 mm (6 in)	600	(0.8)	40	(1.0)

* From **Ventilation of Agricultural Structures.** 1983, ASAE.

- Hose pads off at least once every two months.

- Replace pads once a year or as the manufacturer recommends.

Mist Fogging

The mist fogging method achieves vapour transfer by passing air through a fine water spray. Performance of such systems is not as good as for pad type coolers however the cooling effect can still be as large as 4°C. Such systems are widely used in poultry facilities and are a good low-cost summer cooling option.

Fogging nozzles are mounted on PVC pipe and create a fine mist of water. The small particles of water are easily evaporated - the greater the degree of atomization, the better the system performance.

A variety of pipe configurations have been tried, both inside and outside the barn. The most practical location of the spray nozzles appears to be outside the barn along the inlet vents. Location of the foggers in the inlet air stream takes advantage of the high vapour pressure differential between the dry air and the water droplet and results in higher saturation efficiencies than when foggers are located inside the barn.

A common but inferior method is to string lines of foggers along the inside rafters or trusses of a barn. Evaporation of the water spray is not as efficient since relative humidity inside the barn is

higher than outside. Wetting of litter also occurs with inside mounted mist fogging systems.

Use of mist fogging systems is effective only when RH is less than 75%. **Do not operate when RH is higher than 75%.**

Design of mist fogging coolers

Fogging systems are either low or high pressure and are very simple in construction. Low pressure systems use regular line water pressure whereas high pressure systems include a booster pump to improve automization of the water particles. The extra cost of a booster pump is small; high pressure systems are much more effective and are recommended.

Figure 10.3 shows the suggested mounting arrangement and the components of a high pressure mist fogging system. The misting nozzles should be located so that incoming air passes directly through the water spray. The direction of the spray should be the same as the direction of air flow.

- Boost line pressure at the pump to at least 700 kPa (100 psi).
- Size the system for 12 L/h for each m^3/s air flow (1.5 gal/hr for each 1000 cfm air flow).

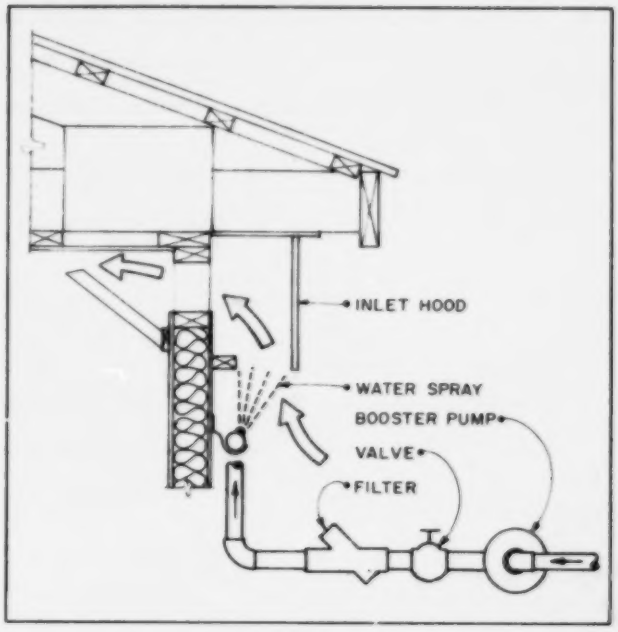


Figure 10.3 Mist fogging system.

- Size pipe for a maximum velocity of 1.5 m/s (5 ft/sec). Usually 1/2 in PVC pipe is used.
- Choose more low volume misting nozzles over fewer high volume misting nozzles.
- If 4 L/h (1 gal/hr) nozzles are used the recommended spacing is 600 mm (2 ft).
- Install an in-line screen filter.
- Add a valve at the end of each line for flushing of the line.
- Use black PVC pipe to reduce algae problems.

- Injection of a 20-30 ppm chlorine solution for 1/2 hr once a month will help control algae and reduce plugging of nozzles.

- Wire thermostat to the pump and a solenoid shut-off valve.

- Nozzle plugging due to hard water scaling can be corrected by treating with a diluted "inhibited muriatic acid" solution for 10-15 minutes.

Roof Wetting

A very simple and inexpensive form of evaporative cooling is to keep the barn roof wet during hot summer periods. This is usually accomplished by stringing a perforated garden drip hose along the roof peak or by mounting garden sprinklers. This method will not solve the evaporative cooling requirements of a barn but it has been used in emergency situations to provide some relief of heat stress when no other form of evaporative cooling was available.

Spray Cooling for Swine

Pigs have very limited ability to sweat and are more susceptible to heat stress than other animals. High temperatures have a significant negative effect on reproductive efficiency and feed conversion. Sprinkling water on the pigs body surface provides effective relief of heat stress.

Most modern swine facilities use sloped concrete floors with a slotted-floor dunging area. Excess water drains into the manure pit. Total waste volume does not generally increase much since spraying is intermittent and pigs drink less when they are kept cool. Spray cooling systems are simple and inexpensive; payback periods are usually one or, at most, two years.

Spray cooling design

Spray cooling nozzles for swine should be located above the dunging area and operate intermittently 2-3 minutes per hour. The best results are obtained with a hollow conical spray pattern. Mount one nozzle per pen about 1800 mm (6 ft) above the slotted floor. Use table 10.2 to assist in nozzle selection.

Table 10.2 Nozzles for Swine Sprinklers*

Pigs per pen	Nozzle size
10	100 L/h (0.45 gpm)
20	200 L/h (0.90 gpm)
30	300 L/h (1.35 gpm)

* From Midwest Plan Service Structures and Environment Handbook. 1983.

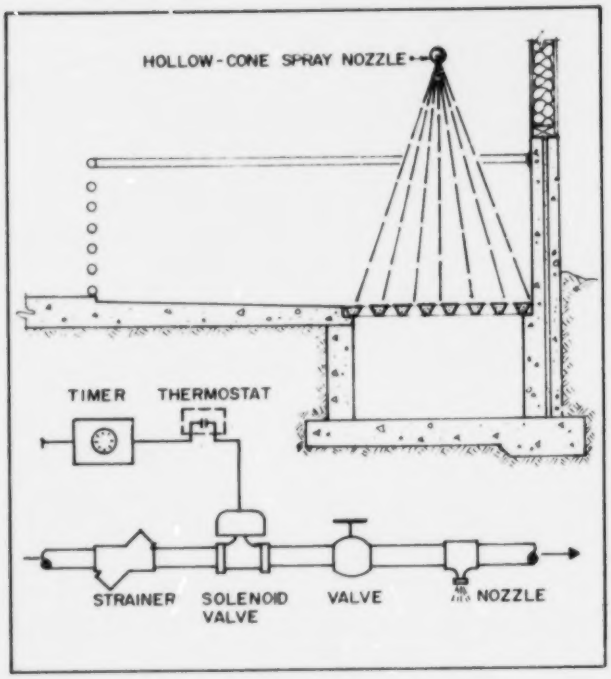


Figure 10.4 Spray cooling for swine.

- Use PVC or galvanized pipe sized for 1.5 m/s (5 ft/sec) maximum velocity.
- Use an in-line filter.
- Select stainless-steel nozzles which provide a coarse droplet conical spray pattern not to exceed the perimeters of the slotted floor area. Avoid wide spray angles which wet the walls instead of the floor - this is a waste of water.
- Use a timer/thermostat circuit which turns the sprinkler on for 2-3 min/hour when temperature rises above the set-point. Four or five 30-second intervals per hour are better than one 2-minute wetting period.
- Observe pig behaviour to determine the optimum thermostat set-point.

CHAPTER 11 HEATING

GENERAL

This chapter provides the information for informed decision making on the type and size of supplemental heating system which is best for the case at hand.

One of the most important considerations is the size of heater. Oversizing the heater is easily as bad as undersizing. An oversized heater will cause large temperature fluctuations as it cycles on and off. Temperature fluctuations put stress on the animal and very often waste energy. **Do not oversize your heating system.**

After the sizing question is answered the next important question is what kind of heat is required: **space heating** is the heating of the entire room; **radiant or spot heating** is the heating of a localized area such as a creep area or small pen (electric heat lamps, gas fired brooding lamps, etc.).

Since radiant heat warms only the receiving surface, it makes sense to use spot heaters in a farrowing creep where it is only necessary to warm the piglets, not the whole barn.

Space heaters should be selected to warm the air as efficiently as possible. Space heaters are usually placed near the air inlets to heat the incoming air as quickly as possible. Preheat hallways are becoming more popular since they reduce cold drafts due to cold outside air being introduced directly into the animal environment.

Another important factor to consider is the management cost of the system in relation to its energy cost. Coal, for example, provides low cost energy but very few farmers would be willing to spend 2 hours every day shoveling coal and stoking the furnace. Similarly, the system should be easy to maintain and keep clean. Dirt and dust build-up is a serious problem for all heating systems.

CALCULATION OF HEATING REQUIREMENTS

The amount of supplemental heat required to maintain a desired inside temperature in a barn is determined by the heat balance relation (equation 11.1).

(Eq. 11.1)

$$q_{sh} = q_v + q_b + q_e - q_s - q_m$$

q_{sh} = supplemental heat required, kW or Btu/hr.

q_v = ventilation heat loss, kW or Btu/hr.

q_b = building heat loss, kW or Btu/hr.

q_e = sensible heat used to evaporate water, kW or Btu/hr.

q_s = sensible heat production of animals, kW or Btu/hr.

q_m = mechanical heat from lights, motors, etc.

Most heat production figures take into account the fact that some of the sensible heat actually produced by the animals is used to evaporate water in the building. The mechanical heat component, q_m, is usually small compared to the other terms and may be neglected. Consequently, equation 11.1 may be reduced to the simpler relation, equation 11.2.

(Eq. 11.2)

$$q_{sh} = q_v + q_b - q_s$$

in which,

$$q_v = (Q/v_s)(h_i - h_o) \quad \text{(Eq. 2.15)}$$
$$= (Q/v_s)c_p(t_i - t_o)$$

$$q_b = (A/R)(t_i - t_o)$$

$$q_s = (\#\ \text{animals})(\text{BSHL})$$

Q = ventilation rate, m^3/s or ft^3/hr

v_s = specific volume of air, m^3/kg or ft^3/lb

$h_i - h_o$ = difference in enthalpy between inside and outside air, kJ/kg or Btu/lb

c_p = specific heat of air, kJ/kg-°C or Btu/lb-°F

= 1000 J/kg-°C approximately

$t_i - t_o$ = difference in temperature between inside and outside, °C or °F

BSHL = Building sensible heat load, W or Btu/hr

(Use table 4.1 to determine BSHL.)

Example:

t_i = 25°C; RH_i = 80%; t_o = -10°C

500 pig weaner barn: 10 kg pigs (average)

building heat loss factor (A/R)

= 103.0 W/°C

- q_s = (500 pigs)(10 kg/pig)(3.1 W/kg)
  = 15,500 W

- q_b = (103.0 W/°C)(25-(-10°C))
  = 3,605 W

- min. ventilation rate
  = (500 pigs)(0.9 L/s/pig) (table 4.1)
  = 450 L/s = 0.45 m^3/s

- q_v = ((0.45 m^3/s)/0.87 m^3/kg) x (25-(-10°C))(1000 J/kg-°C)
  = 18,100 W

  (See example design problem number 3.)

- q_{sh} = 18,100 + 3,605 - 15,500
  = 6,205 W

So in the above example 6.2 kW of supplemental heat are required. But such calculations are sensitive to the factors of stocking density, assumed inside and outside temperatures, and the calculated building heat loss. To properly size the heating system it is necessary to take the different cases which might occur. For example:

Lower stocking density

If there are 400, 10 kg pigs in the barn instead of 500 then:

- q_{sh} = 18,100 + 3,605 - 12,400
  = 9,305 W

Lower inside temperature

- If t_i = 20°C only 3.5 kW of supplemental heat is required (but such a low temperature would not be advisable).

By selecting the worst case data for stocking density, winter ventilation rate, and minimum winter outside temperature, equation 11.2 provides an estimation of **the maximum instantaneous heating demand.**

SELECTION OF HEATING SYSTEMS

In order to proceed logically in the heating system selection process it may be helpful to answer the following list of questions:

1) What is the maximum instantaneous heating demand?

2) What are the specifics of the heating requirements?
   - Type of animal or poultry
   - Is spot heating necessary?
   - Is space heating necessary?
   - Estimate of annual heat energy requirements

3) What source of energy is least costly in the location in question?
   - Estimate energy costs using different fuels

4) What are the capital costs of various suitable heating systems?

5) What are the total costs (i.e., capital + operating cost) of the various suitable heating systems and which is cheapest?

6) What amount of management is necessary for each of the systems under consideration?

- Ease of cleaning, adjustment, visual inspections
- Frequency of maintenance
- Fuel handling and management requirements

7) How closely will the system come to providing an optimum thermal environment for the animal or bird?

8) Based on all of the above which is the best overall option?

HEATING SYSTEM TYPES

(See References for more details on the systems described below.)

Hot Water Heaters

Hot water is a common type of supplemental heating method for high heat requirement facilities such as Hog, Dairy and Poultry buildings. It is generally considered economical and practical only where the heating demand is 20 kW (70,000 Btu/hr) or more.

Advantages

- Minimum disturbance of air flow circulation patterns; compatible with most ventilation systems.
- Generally very energy efficient.
- Low maintenance requirements; easy to clean, very reliable.
- Because no air filtration for dust removal is required, hot water heat is safer and more trouble free: no clogged air filters, fire hazard or other dust related problems.
- For large buildings having several rooms (or heating zones), hot water provides excellent individual zone control from a central boiler.
- Quiet operation.

Disadvantages

- High capital cost; especially for small or one room buildings.
- Does not provide spot heating.
- Generally slower response time to a change in heating demand than a forced air or radiation system.

A hot water heating system consists of three major components:

1) The water boiler and pump
2) The radiator (pipe network)
3) The control system

Figure 11.1 shows a typical boiler and pump arrangement.

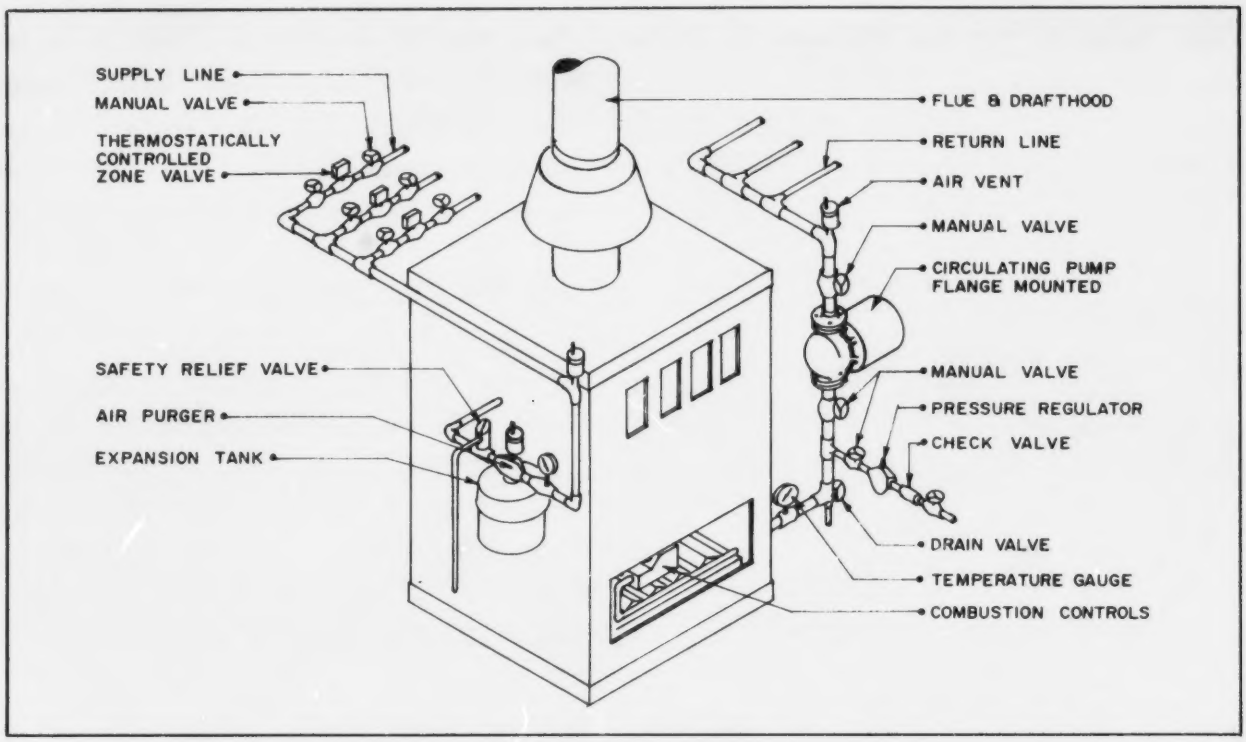


Figure 11.1 Hot water heating boiler.

Radiators are wide and varied in design. There are far too many different types of radiators to list here, however three examples are shown in figure 11.2. The objective of any radiator is to optimize heat transfer from the water to the space being heated; all radiator designs attempt to achieve heat transfer while at the same time satisfying other design constraints.

A primary concern in livestock facilities is dust and dirt build-up. Dirt collecting on the heat transfer surface will act as an insulation and seriously reduce radiator effectiveness. Designs that are easy to clean and which are resistant to dirt build-up are best. **Finned-tube convectors** like the one shown in figure 11.2 are ineffective in a dirty or dusty environment and are not recommended except in dust and dirt free locations.

For most applications the **black steel pipe** option shown in figure 11.2 (or some alternate pipe configuration) is preferable. It is sturdy, easy to clean and is not badly affected by dust. Its disadvantages are that more pipe is required (since there is less surface area exposed per unit length than for fin tube) and labour costs associated with installation are higher. Black steel pipe radiators may consist of one or more pipes hung on

wall mounted sliding brackets to allow for pipe expansion.

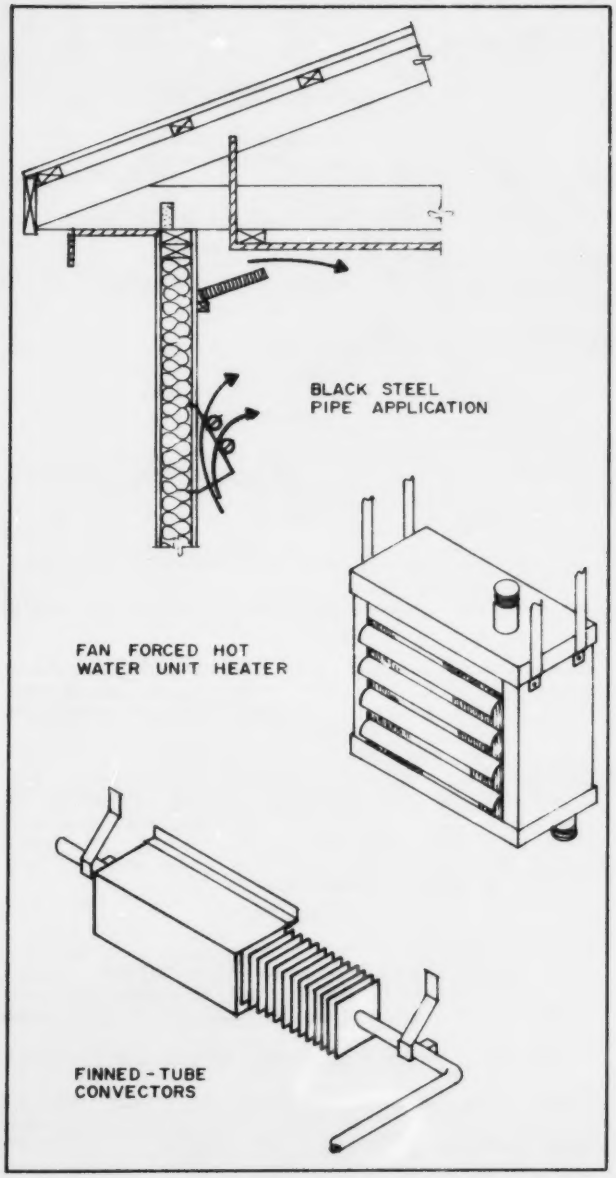


Figure 11.2 Three different radiator types.

Hot water unit heaters are easily clogged by dust and should be inspected and cleaned frequently. They are ideal for small space heating applications such as shops, milk rooms, calf pens, etc. Other radiator systems include, forced air hot water brooders, radiant pipe hot water heaters and brooders, and in-floor heating.

Warm Floor Heating

Either hot water pipes or electric resistance cables can be imbedded in the concrete floor (or below it) to provide floor heat. Such systems conduct and radiate heat to the air and livestock.

Advantages

- Provides animal comfort in heated areas.
- Dries wet floors quickly; melts ice.
- Uniform heating.
- Floors have a large heat capacity; floors retain heat for long period of time.

Disadvantages

- Pipes can be damaged if the floor cracks and repair is expensive.
- Response to changes in heating load is very slow; warm up time may be as much as 2 days, thereby increasing fuel costs.

- System may not meet maximum instantaneous heating demand; i.e., system may not be able to keep up during winter minimum temperatures.

In addition to the above advantages and disadvantages, those of the **Hot Water** section also apply.

Floor heat is most effective in maintaining a warm floor in farrowing creeps, weaner pens, milking parlour pits, and farm shops. Use of floor heat to meet space heating requirements may be limited because of the following reasons: first, the floor temperature required for effective space heating may be higher than 30°C (86°F) which is the upper limit for comfort; second, high floor temperatures cause increased heat losses to the ground below the floor; and third, the whole floor area must be heated for effective space heating.

The most serious disadvantage to floor heat is that response time to sudden changes in heating demand is slow. It may be necessary to install a supplementary space heating system for quick response.

Gas Fired Heaters

Gas fired heaters are widely used in agriculture in a variety of applications. Natural gas or propane is the heating fuel of choice for a range of systems, however in this section **gas hover brooders** are mainly referred to.

Advantages

- Provides spot heating for chicks, young pigs, calves and lambs.
- Use of inexpensive energy; good energy efficiency.

Disadvantages

- Extreme temperature variation occurs as chicks (or pigs) move out of the sphere of influence of the brooder.
- Extra moisture is produced as a by-product of combustion which must be vented (or ventilation rate increased).
- If improperly adjusted can produce toxic carbon monoxide gases.

Brooders are designed to supply radiant heat on the backs of the chicks. The burners on many brooders are formed in a ring to provide a larger radiant area and a larger warm comfort zone.

Other types of gas fired heaters include **fan forced gas hot air unit, overhead gas infrared**, and any number of central gas furnaces which provide circulating air or hot water heat.

Gas and moisture production

When natural gas (CH_4) burns, the by-products of combustion are CO_2, H_2O and to a limited extent, CO. If the burners are unvented, such gases and moisture is added to the room environment. Ventilation rate may need to be increased to remove these gases. The chemical description of the combustion of natural gas is as follows:

Reaction path 1:

$$CH_4 + 2O_2 = CO_2 + 2H_2O + 890\ kJ$$

Reaction path 2:

$$CH_4 + 2O_2 = CO + 2H_2O + 1/2O_2 + 607\ kJ$$

When burners are properly adjusted, the flame burns cleanly and combustion follows path 1; no CO is produced.

When burners are improperly adjusted, combustion is incomplete and part of the combustion follows reaction path 2; CO is produced and should be removed from the environment by an approved system of venting.

In either case, moisture is always produced by unvented gas-fired burners. Such moisture production should be added to animal moisture production figures to determine minimum ventilation rate for moisture removal.

Gas and moisture production may be estimated from the total heat output of a gas fired burner. Equations 11.3, 11.4 and 11.5 may be used for estimating water, CO_2 and CO production from a natural gas furnace.

H_2O production:

SI:

$$MP = 9.2 \frac{\text{Furnace output}}{E} \qquad \text{(Eq. 11.3)}$$

Imperial:

$$MP = \frac{\text{Furnace output}}{175{,}000\ E}$$

MP = moisture production, g/h or lb/hr.

Furnace output = heat output of furnace kW or Btu/hr.

E = efficiency of furnace (usually between 0.7 and 0.8)

CO_2 production:

SI:

$$CO_2 = 11.3 \frac{\text{Furnace output}}{E} \qquad \text{(Eq. 11.4)}$$

Imperial:

$$CO_2 = \frac{\text{Furnace output}}{140{,}000\ E}$$

CO_2 = carbon dioxide production, g/h or lb/hr.

CO production*:

SI:

$$CO = 0.1 \frac{\text{Furnace output}}{E} \qquad \text{(Eq. 11.5)}$$

Imperial:

$$CO = \frac{\text{Furnace output}}{3,100\ E}$$

CO = carbon monoxide production g/h or grains/hr (7000 grains = 1 lb)

* Based on a poorly adjusted burner - CO production could be higher than that predicted by equation 11.5.

Gas Infrared Heaters

The gas infrared concept is based on radiant, rather than space air heating. The capital cost of such systems are generally very high, but the advantages of gas infrared heating are leading to its increased popularity.

Advantages:

- High efficiency (approx. 90% combustion efficiency).
- No heater induced air drafts.
- Fast response to changes in heating demand.
- Quiet operation.
- Low maintance and cleaning requirements.
- Radiant heating is greatest directly underneath and decreases with distance from the overhead radiator; chicks or animals are permitted to find the most comfortable temperature zone.

Disadvantages:

- High capital cost.
- May not be suitable for low ceiling applications.

The gas infrared system consists of a series of small burners connected by 100 mm black iron pipe, suspended from the ceiling and immediately under a bright metal surface reflector. The pipe and burner system is kept under vacuum by a fan at the discharge end and fresh combustion air is brought in from the outside of the barn.

The temperature of the black pipe is kept at a maximum of 480°C (900°F) and does not glow red. The black pipe surface emits long wave radiation in the infrared spectrum. At this temperature, the black iron pipe has emmissive properties which approximate the perfect black body emitter; radiant heat energy spreads out

over the ground surface below the pipe network and warms all objects it strikes. Air temperatures in gas infrared heated barns are usually slightly less than for conventionally heated spaces. This decreased air temperature is usually not a problem since the increased animal convective heat losses are balanced by increased radiation heat gains.

Gas infrared systems are suitable for brooder operations and weaner-grower swine barns. The gas infrared concept is best suited to heating large flat areas.

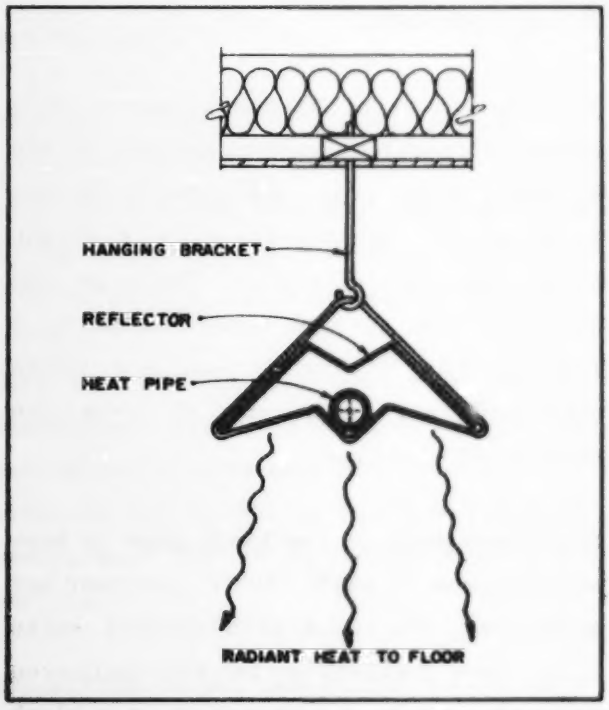


Figure 11.3 Co-Ray-Vac radiant gas heating system.

Electric Infrared Heaters

Electric infrared heaters are widely used for spot heating requirements. Electric heat lamps are used for piglets, calves, chicks and lambs. Capital costs of such lamps are low and installation is relatively easy, accounting for much of their popularity. Since they radiate heat only where it is needed and heat very little air, they conserve energy.

Advantages

- Provides spot heating exactly where required.
- Are inexpensive to purchase and install.
- Light from brooder lamp attracts small pigs and chicks, teaching them where to find warmth.

Disadvantages:

- Safety has been a problem - but with proper installation safe operation is now possible.
- Electricity is one of the more expensive energy sources.

There are three main types of electric infrared heating elements: heat lamps (like those in your bathroom), quarts or

silica tube, and metal sheath elements. Heat lamps are generally used for smaller load applications such as farrowing and weanling pens, while the quartz or silica tube and metal sheath element heaters are used for higher load applications such as calf warming.

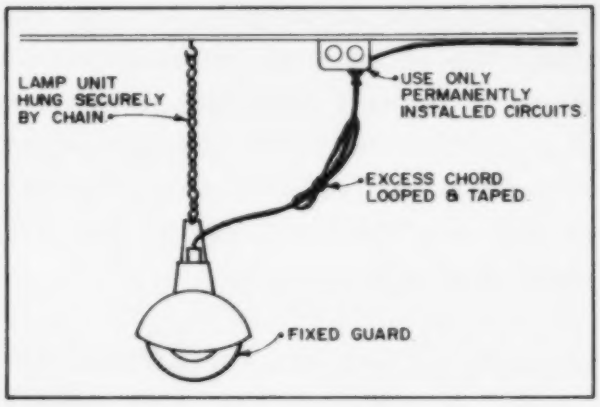


Figure 11.4 Suspended spot heat lamp.

For larger load applications the quartz or silica tube lamps should be selected over the metal sheath type of heater as they are more energy efficient and have a faster heat-up time.

Dimmer switches

Farmers commonly raise or lower heat lamps in order to achieve desired temperatures at the animal's back. This technique wastes energy since it reduces temperature by increasing the area being radiated by the lamp and heat falls outside the needed area.

A better method of temperature control is to change the heat output of the lamp either by selecting a different bulb size or by installing a dimmer switch.

A simple dimmer switch reduces lamp power consumption, increases lamp life and eliminates the need for frequent adjustment of lamp height or bulb size.

For up to 600 W installations (usually two lamps) common household dimmer switches may be used. For larger capacity installations industrial quality switches are necessary.

INTERLOCKED HEATING CONTROL

One of the problems with heating systems (from an energy conservation point of view) is overventilation while the heater is on. If ventilation fans are running while the heater is on, heat is being produced by the heater and pumped right out of the building by the ventilation system.

The usual technique for preventing energy waste is installing thermostatic controllers on both the heating and ventilating equipment. The sizing of equipment and the setting and sensitivity of the thermostats can, however, lead to a problem. If a large capacity fan starts during a warm period, the room temperature may be drawn below the set-point of the

thermostat for the heating system. The heating system then cuts in and maintains the room temperature above the fan shut off temperature. This leads to both units operating at the same time. The result is a situation where the fan and heater are pumping energy right back out of the building. This is a very expensive waste of heat energy.

Sometimes the problem arises out of thermostat sensitivity and it may be very hard to correct. There are three possible solutions.

1) Adjust the set-points for the fan thermostats several degrees above the heater thermostat set-point.

2) Install more sensitive thermostats.

3) Install interlocked heating/ventilation controls.

Options 1) and 2) may not solve the problem so it may be necessary to install electrically interlocked heating/ventilation controls.

Canada Plan Service publishes an electrical schematic for an interlocked control (CPS Plan 9701). Any good electrician can install this system, or a similar system, to meet the specific needs of the barn ventilation control system.

The electrically interlocked heating/ventilation system prevents simultaneous operation of the heater and "greater than minimum rate" ventilation fans.

CHAPTER 12 TROUBLE SHOOTING

This section is perhaps the most important part of the handbook. While the ideal case is always to design a new ventilation system, there are many existing facilities in need of improvement. The field serviceman must be able to identify problems in a ventilation system and determine corrective measures.

The purpose of this chapter is to present guidelines for the producer, fieldman and designer when trouble-shooting a farm ventilation system.

UPON ENTERING THE BARN

The most important skills a trouble-shooter can have are powers of observation. **Be observant** and take notes of your observations.

Behaviour of Animals

Take note of animals with respect to their penning arrangement. The livestock and poultry will often give clues to the nature of the ventilation problem. For example:

- Pigs are huddling in groups; **room is probably too cold.**
- Chicks are grouped under the brooder; **room is too cold.**
- Chicks are grouped near doorways or cracks in the walls; **room is probably too hot.**

Odour

The human sense of smell is quickly fatigued when subjected to strong odour. Identify odours present in the barn in the first minutes of your inspection.

- **Ammonia (NH_3):** Strong odour of ammonia indicates insufficient ventilation. Ventilation rate may need to be increased. Ammonia is produced by the degredation of manure - look at the manure management practise. Are the floors wet? Is the problem likely one of ventilation or manure management? Are the manure pits ventilated?
- **Hydrogen Sulfide (H_2S):** A strong "rotten egg" smell indicates elevated levels of hydrogen sulfide. Ventilation may again need to be increased. Manure management could also be the problem.

Dust

Heavy dust loading in the air indicates either:

- relative humidity is too low: dry air is causing increased dust formation; or
- Inadequate ventilation: dust is not being removed.

Cleanliness of pens

Poor ventilation can cause behavioural problems in animal dunging habits. Swine are especially sensitive to ventilation, and poor air circulation patterns can lead to poor dunging habits.

DON'T EXPECT THE VENTILATION SYSTEM TO OVERCOME BAD ANIMAL MANAGEMENT PRACTICES!

Humidity

Is the room too damp? Check walls and floors for signs of condensation. Is the litter moist or dry? Does the air feel damp or "close"?

MEASUREMENTS

The subjective observations mentioned above need to be investigated rationally. Measurements of the barn environment need to be made in order to isolate ventilation problems.

Temperature

An accurate thermometer is essential for evaluation of a controlled environment barn. Accuracy may be checked by placing the thermometer in a 50/50 mixture of clean ice and water which has sat for about 30 minutes. The temperature of the water will be exactly 0°C or 32°F; the thermometer should read the same.

Thermometers must be kept **clean**. Dirty thermometers will have a slow response to changes in temperature.

Thermostat thermometers are very often unreliable. All thermostats should have an additional **tested** thermometer installed beside them for more accurate readings. Digital thermometers give more accurate readings and a faster response time. They are, however, much more expensive. A good quality mercury-in-glass thermometer is better since they are inexpensive and several can be installed to monitor temperatures in all barn locations.

Hi-Low Thermometers are extremely valuable measuring tools. Temperature fluctuations put stress on livestock. A hi-low thermometer will read present temperature and the maximum and minimum temperatures which occured since the last thermometer reset.

Large temperature fluctuations (>3°C) can indicate:

- poor quality thermostats - not sensitive enough;
- thermostat settings incorrect - too high or too low;
- location of thermostats gives wrong temperature readings - thermostat is in a cold or hot spot or is not shielded from radiation heat gains or losses;
- thermostats are dusty or damaged;
- fans are oversized;
- heaters are improperly controlled or sized.

Thermostat location and installation is important. The objective is to place the thermostat in a location which is representative of the air temperature at animal level. Practical considerations dictate the following guidelines:

1. Thermostats should be as close to animal level as possible but still out of animal reach.
2. Thermostats should not be located on outside walls; cold outside temperatures will adversely affect thermostats placed on outside walls.
3. Thermostats should be shielded from radiation (windows, heaters and lamps).
4. Thermostats should not be located near doors or other frequently used openings.

Relative Humidity

A **sling psychrometer** is used to measure relative humidity. Two thermometers, one with a wetted wick over the bulb and one without, measure wet and dry bulb temperatures respectively. Relative humidity is then obtained from a psychrometric chart. If used properly the sling psychrometer gives accurate results and can be used to check the readings of **hygrometers**. Hygrometers measure relative humidity continuously and may be placed anywhere within a building.

Hygrothermograph

A hygrothermograph measures temperature and relative humidity continuously. They are placed in one location and record continuously for as much as seven days.

Because relative humidity is measured by a hygrometer, the instrument should be checked periodically with a sling psychrometer. Dust build-up can render the hygrometer inaccurate. RH readings are,

in any case, accurate only to plus or minus 5%.

The hygrothermograph is a very valuable diagnostic tool. Temperature and relative humidity fluctuations can be continuously monitored.

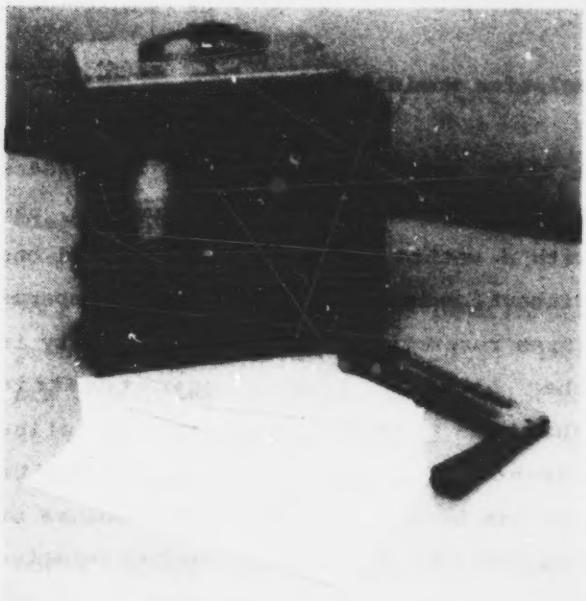


Figure 12.1 A Hygrothermograph with sling psychrometer.

Figure 12.1 shows a hygrothermograph and Figure 12.2 shows two sets of chart readings. The top set of chart readings is of a well controlled barn. Note that temperature fluctuation due to fan cycling is less than 2°C and relative humidity fluctuation is consistently in the range 58% to 72%.

The bottom set of chart readings is of a poorly controlled barn. Temperature and relative humidity fluctuations due to fan cycling are much too large and rapid. This barn's control system needs to be improved.

Smoke Testing

A burning pipe or cigarette can give an indication of air flow patterns but there are much better devices available. The Draeger air current detection kit or a propane-fired insect fogger (with light mineral oil instead of insecticide) may be used.

Smoke testing shows properly or improperly adjusted air inlets, undesirable leaks around doors, flow patterns in pens, and room circulation patterns.

Do not fill the room with smoke. Short bursts of smoke should be released in suspect areas and the leading edge of the cloud observed. Experience will greatly improve interpretation of the results.

Air Velocity

Hand-held velometers are available for measuring air velocity. Observations from the smoke test can be transformed into hard numbers. Measurements should be taken at livestock level and at air inlets to determine air speeds.

Air inlet velocity should be checked.

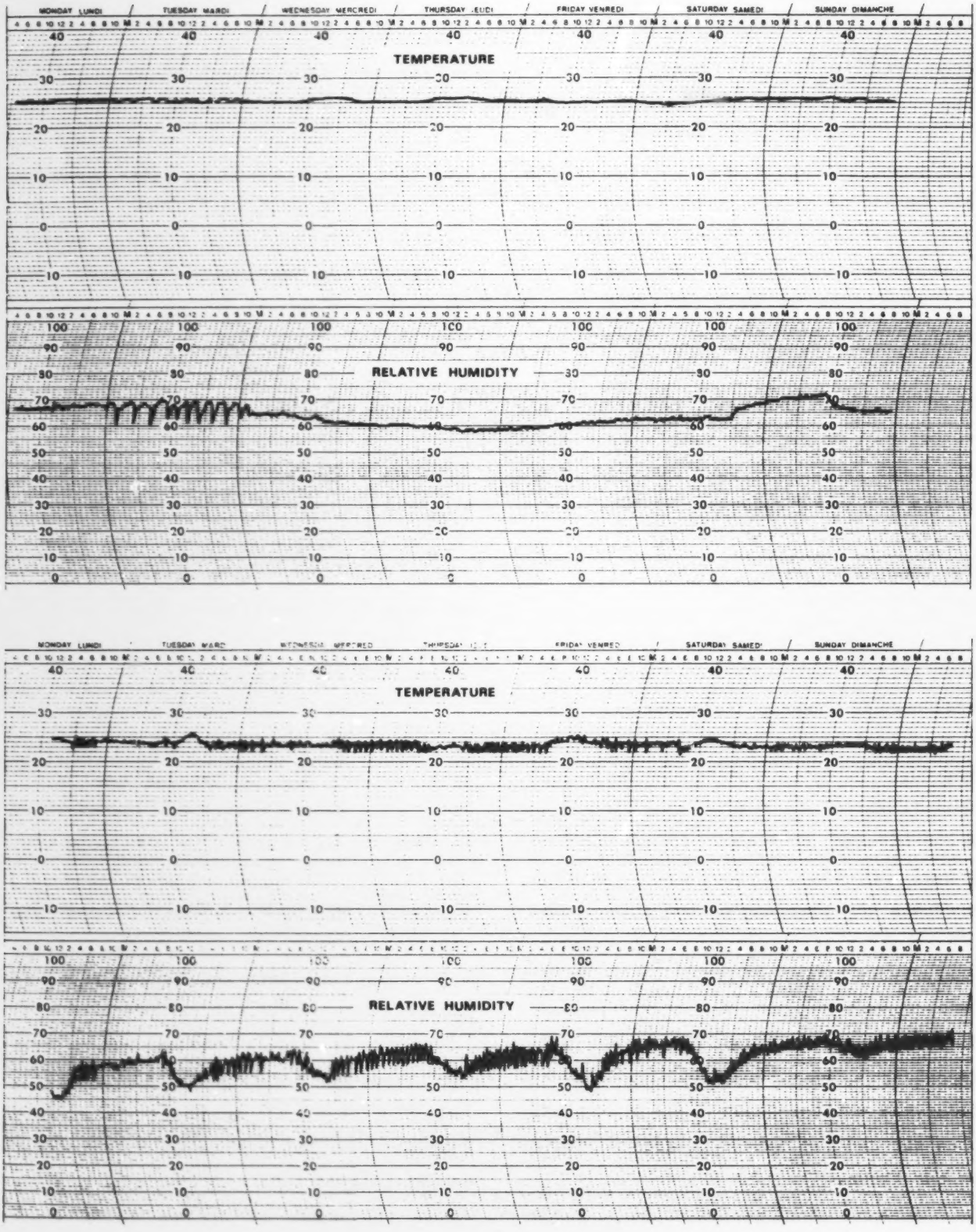


Figure 12.2 Example temperature and relative humidity chart records. The top set of chart records indicate a well controlled barn. Note the large temperature and relative humidity fluctuations in the bottom set of chart records. Also notice that RH fluctuations correspond to temperature fluctuations.

Most air inlets are designed for air speeds of 4 to 6 m/s (800 to 1200 ft/min).

For young livestock sensitive to drafts, air speeds at animal level should be less than 0.2 m/s (40 ft/min). High air speeds reduce the effective temperature the livestock experiences due to the action of convective heat transfer.

Your hand can be used to detect air velocity. If you can feel a cold draft at animal level with your hand, it is too drafty for young animals.

Gas Concentration

Levels of gas contaminants can be measured with various instruments. The section in chapter 3, Animal Physiology, gives recommended levels for CO_2, H_2S, NH_3, CO, and CH_4.

Colourimetric gas tubes (e.g., Draeger tubes) provide a fast and relatively accurate gas sampling method for contaminating gases at a cost of about $3.00 per analysis.

When testing for CO_2 and H_2S it is important to take measurements near the floor, as both of these gases are heavier than air and may "pool" in manure pits or on the floor.

If there is a high concentration of water droplets or dust in the air, the colourimetric gas tube method of sampling may underestimate NH_3, CO and H_2S levels. All of these gases are water soluble and the gas tube will not measure gases dissolved in water or dust particles. If doubt exists, other more sophisticated methods of analysis may be necessary. The Engineering Branch of the Ministry of Agriculture and Food is interested in this problem and may be able to provide you with some assistance.

Pressure Differential

The static pressure differential between the inside and outside of a barn can be measured with a static pressure guage or a manometer. The Dwyer manometer is often used for this purpose.

Exhaust fans create a vacuum in the room which drives air in through the inlet. Too little vacuum causes the incoming air to flow lazily, mix poorly with room air, and fall to the livestock level as a cold draft. Too much vacuum reduces the ventilation rate, burns out fan motors, and causes high air speeds resulting in drafts.

The manometer is very useful for adjusting inlet settings. Response is nearly instaneous. Static pressure differential varies from barn to barn but will usually be in the order of 2.0 mm (0.08") of water column. This value is a good starting point for inlet adjustment.

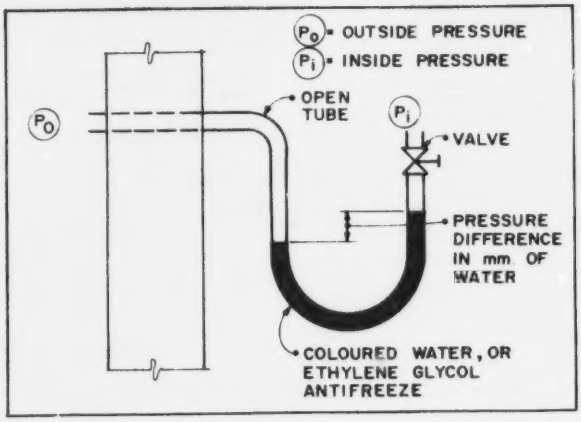


Figure 12.3 A simple water column differential pressure manometer.

ANALYSIS OF FANS, INLETS AND CONTROLS

Fans

In order to evaluate the system, fan design and installation needs to be checked. The following checklist should be used.

1. Number and size of fans:
   a) Model number, location and total number of fans;
   b) Horsepower, (hp);
   c) Rated speed, RPM;
   d) Blade diameter;
   e) Blade configuration;
   f) Sheave sizes for belt drive; or
   g) Single, two or variable speed adjustment.
2. Shutters and hoods in place and functioning.
3. Cleanliness - dust build-up on blades, shutters and hoods.
4. Adequate and correct wiring:
   a) Motors overload if voltage drop is too high;
   b) Fans will run backwards if wired incorrectly.
   c) Fan stages can easily get mixed up to run off/high/low instead of off/low/high.
5. Maintenance: check bearings, belts and vibration.
6. Location of blade assembly with respect to housing.
   Fan blade should be 2/3 inside the bell housing with 1/3 projecting through the orifice on the discharge side.

After going through the checklist the system design can be evaluated and any deficiencies in the equipment or maintenance can be corrected (points 2 through 6 in checklist). The fan sizes and capacity can be checked using the procedures given in chapter 4.

Controls

The control system should also be evaluated with a checklist.

1. Number, type and location of controls:
   a) Model numbers;
   b) Map of control locations.

2. Control settings - all locations.

3. Cleanliness - are thermostats and instruments free of dust and dirt.

4. Quality of instruments - what are the specifications of the thermostats - are they good enough? (See chapter 7.)

5. Flexibility of control:
   a) What is the minimum ventilation rate?
   b) What is the maximum ventilation rate?
   c) Do minimum and maximum correspond to required winter and summer ventilation rates?
   d) Does the system provide enough fan stages?

6. Variable fan speed controls should be set between 20% and 100% of full rated speed (e.g.: 320-1600 rpm):
   a) Continuously running fans at less than 20% of full rated speed may lead to premature burn out of fan motors;
   b) greater than 20% settings assure air movement through the fan housing;
   c) adjustment of the minimum speed is by front mounted control or by set screw on the back of the unit in the case of 'Ranco' products.

The adequacy of the control system may then be evaluated on the basis of measurements of system performance.

Air Inlets

Checklist

1. Capacity and adjustment:
   Can area be adjusted to obtain minimum and maximum ventilation rates with a constant inlet air velocity of 5 m/s (1000 ft/min).

$$A = \frac{Q}{VC_d} \qquad \text{(Eq. 12.1)}$$

A = inlet area, m^2 or ft^2
Q = ventilation rate, m^3/s or ft^3/min
V = air velocity in inlet, m/s or ft/min
= 5 m/s +- 1 m/s (1000 ft/min +- 200 ft/min)
C_d = coefficient of discharge (use C_d = 0.7 if no more reliable value is available)

2. Inlet air velocity:
   Should be within 5 m/s +- 1 m/s (1000 ft/min +- 200 ft/min).

3. Inlet configuration:
   a) centre inlet, poly-tube system, etc.
   b) inlets on both sides of the barn are recommended for buildings up to 12,000 mm (40 ft) in width.

4. Inlet maintenance:
   Curtain type inlets must be in good repair (old fabric is affected by dust and moisture and may need repair or replacement).

5. Direction of air flow - use smoke test
   a) to front of animals for tethered sows or calves (draft on rear end of tethered animals is **undesirable**);
   b) to litter alleys and gutters of animals in group pens - the dunging area should be slightly uncomfortable to encourage resting in more comfortable bedding or resting areas.

6. Winter operations - Air mixing and distribution:
   a) watch for low air flows which do not induce adequate air mixing or distribution (cold drafts, temperature stratification, hot spots, etc.);
   b) If inlet setting is manual, set inlets to the opening which corresponds to the minimum winter ventilation rate;
   c) See figures 12.4, 12.5 and 12.6 for possible solutions to air mixing problems. Use stirring fans. Poly-tubes are also a possible solution to inadequate air mixing.

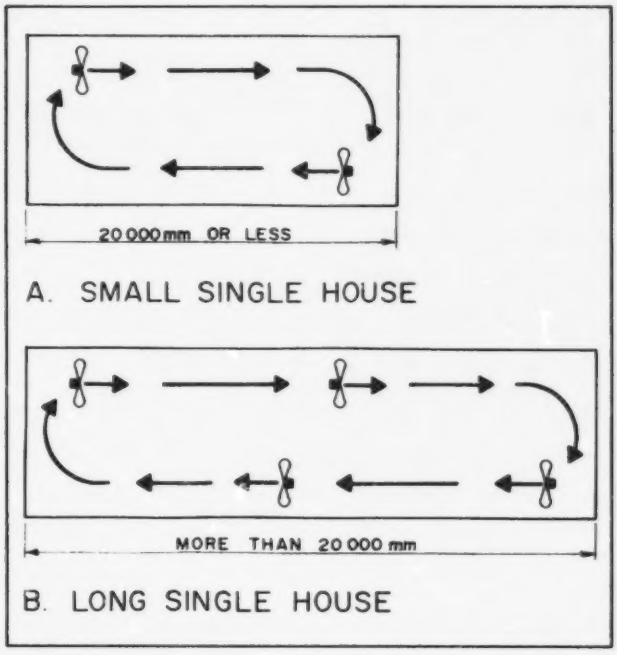


Figure 12.4 Suspended mixing fans. Fans are hung from the ceiling and circulate air around the barn.

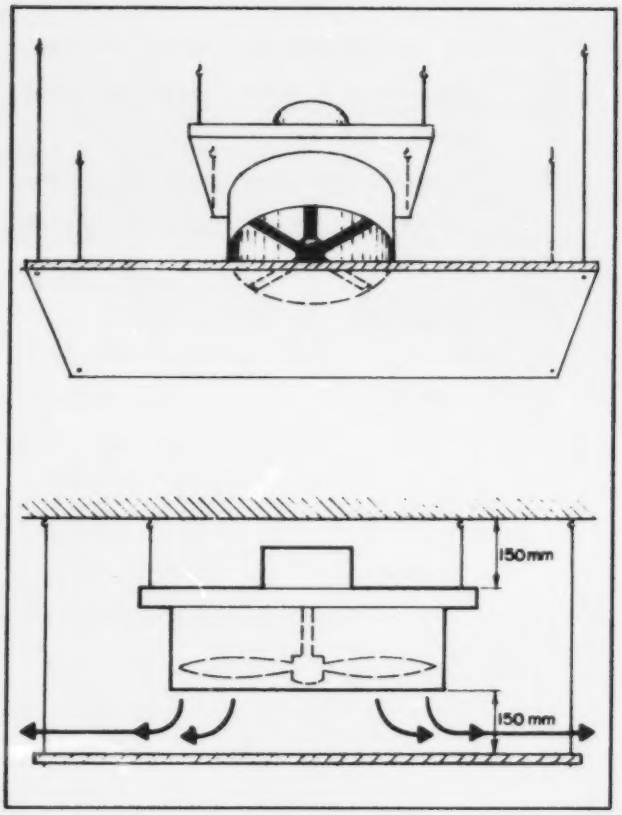


Figure 12.5 Suspended stirring fan or blender.

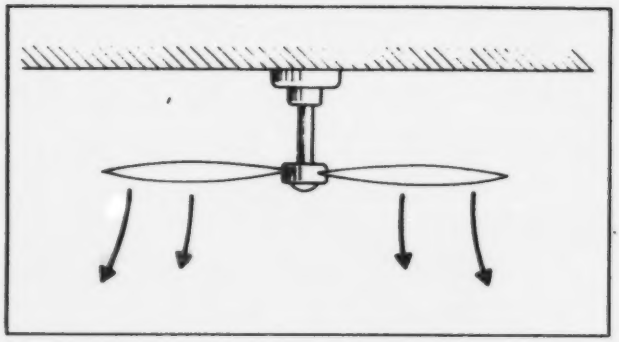


Figure 12.6 Large diameter ceiling fan. Best application with ceilings greater than 3000 mm (10 ft) in height.

TROUBLE-SHOOTING GUIDE (Problems and possible causes)

1. Room is too cold

- Thermostats on fan controls are set too low.
- Thermostats are not properly calibrated - mount good thermometers beside thermostats to check calibration.
- Thermostat is located too close to a heater or ceiling and turns on exhaust fan too soon.
- Air inlet controls malfunctioning.
- Air inlets open too wide; little or no static pressure.
- Poly-tube system is oversized and is providing too much ventilation.
- Air recirculation system is causing high air speeds (drafts) which are chilling the animals.
- Inadequate insulation in walls and ceilings; R-value inadequate or poor vapour barrier causing moisture build-up in walls.
- Cracks in construction are causing heat losses (windows, doors, summer fan openings).
- Animal stocking density too low; animal heat production is less than building heat loss.
- Supplemental heaters are too small or are set too low.
- Location of heaters is poor for quick heating of incoming air.

2. Room is too warm

- Thermostats on fan controls are set too high.
- Thermostats are not properly calibrated; mount good thermometers beside thermostats to check calibration.
- Thermostat is close to an air inlet or cold surface and turns off exhaust fan too soon.
- Air inlet controls malfunctioning.
- Air inlets are too small or clogged; check screens in soffit; static pressure is too high (are doors hard to open from the outside?).
- Fans are not well maintained; **cleaning and maintenance is required once a month.**
- Fan capacity is insufficient.
- Exhaust fans are located too close to inlets and air is short circuting.
- Supplemental heaters are too large or are set too high.

3. Air is damp - Relative humidity is too high

- Not enough supplemental heat (is room too cold also? Go to 1).
- Inadequate ventilation (is room too warm also? Go to 2).
- Thermostat is not set or calibrated properly.

- Fans need maintenance and cleaning once a month.
- Exhaust air is being reintroduced into a nearby inlet.
- Insulation is inadequate; vapour barrier missing or installed on wrong side.
- Manure handling is poor (are pens clean? Do you smell ammonia?).
- Watering devices are leaking.
- Livestock have scours.
- Gas heaters are unvented and water of combustion is being added to the air.

4. Condensation is a Problem

- Insulation is inadequate: single glazed windows? Rodents eating insulation?
- Weather: excessive cold; problem may correct itself when weather warms up (cold weather is causing low level ventilation; e.g. prevailing winds are making North and East walls colder).
- Poor air circulation is causing dead air space on walls.
- Ventilation rate is too low.
- Part of barn is unoccupied, creating a cold spot and leading to condensation in this area.

5. Barn is drafty (cold and hot spots)

- Air inlets open too much; static pressure is low and cold air is dropping directly on animals.
- Air inlets open too little; static pressure is high and high velocity air flows are causing undesirable drafts.
- Size of poly-tube unit is too large or turbulence exists between adjacent tubes.
- Ceiling or wall mounted circulation fans are moving too much air.
- Obstruction near air inlet is deflecting air downwards.
- Undesirable inlets (doors, chutes, manure pipes, etc.) exist.
- Air inlet design is poor.
- Inlet opening is irregular (open both too little and too much).
- Exhaust fan is located too close to inlet (open window?) and air is short circuiting.

6. Fans are not working

- Wiring of fans is incorrect; check direction of fan rotation.
- Control wiring is incorrect.
- Control system is malfunctioning.
- Breaker for fan circuit is tripped (check the obvious).

7. Fuel costs are too high

- Control system is not properly adjusted or is malfunctioning.

- Inadequate insulation and barn construction (leaking doors, windows, etc.). Rodents are eating insulation in the walls and on the foundation perimeter.
- Poor or no vapour barrier has led to wet and ineffective insulation.
- Too much ventilation
- Heaters are not located to warm incoming air quickly.
- Install interlocking relay for thermostat/heater control circuit (CPS plan 9701).
- Replace heaters with radiant type heaters if spot heating is all that is required.
- Energy efficiency is a complex problem. Extensive redesign and analysis of both system and management may be necessary.

8. Fogging

- Add supplemental heat.
- Fogging at inlet in cold weather is common; there may be no problem.
- Air-to-air heat exchanger may help. See chapter 9, Heat Exchangers.

9. Air is odorous

- Ventilation is inadequate.
- Minimum ventilation rate is too low; minimum ventilation rate should be increased to control odour, rather than moisture.
- Thermostat is close to an air inlet or cold surface and turns off exhaust fans too soon.
- No manure pit ventilation system exists.
- Poor quality air is entering room from an odour source (e.g. manure storage or pit fan).
- Manure management system is poor.

10. Temperature is fluctuating

- Fans are too large (ventilation steps are too large).
- There are no fan hoods (or they are improperly mounted) for wind protection. This is especially a problem where variable speed fans are used.
- Heaters are too large or are set too high.
- Fluctuating daily weather conditions in Spring and Fall.
- Thermostat is set too low.
- Thermostat operates fans incorrectly:
  - i) incorrect wiring causes a two speed fan to cycle off /high/low rather than off /low/high or to operate on only one speed.
  - ii) thermostat is off calibration.
  - iii) thermostat is dirty.
  - iv) thermostat is of poor

quality; on/off differential is too wide (> +- 1°C).

v) two-speed thermostat settings are too close so fans operate primarily on high.

vi) one thermostat operates too many fans.

vii) Fans and heaters are not interlocked.

- Thermostat is too close to a heater or ceiling and turns off exhaust fans too soon.
- Use of cycle timer. Remove timer and install smaller fan or fans.

11. Swine: pens are dirty*

- Clean in morning, dirty as day progresses.
  - i) temperature is fluctuating (check with hi-low thermometer).
  - ii) circulation pattern is different at different ventilation rates.
- Dirty in summer, clean in winter.
  - i) summer/winter circulation patterns are different; check inlet design and setting.
  - ii) ventilation rate is too low, conditions are drafty
  - iii) try spray cooling.
- Dirty all the time.
  - i) pens are drafty; correct circulation problems.
- Some dirty pens.
  - i) draw floor plan and perform smoke tests to determine air flow patterns in affected stalls and attempt to correct.

* Problem may not be just one of ventilation: refer to **Developing Good Dunging Habits of Swine on Partially Slotted Floors**, B.C. Ministry of Agriculture and Food, publication number 381-610-1.

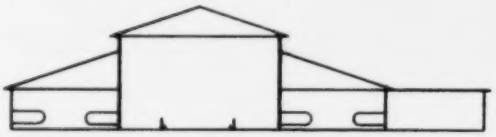
12. Poultry - Common Problems

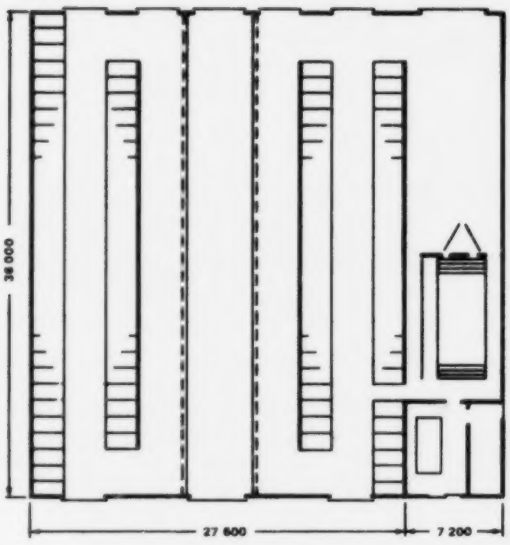
- Operator is under-ventilating the barn to conserve heat.
  - i) check ventilation rates required for bird age and number and modify control settings.
- Poor air mixing.
  - i) inadequate size and number of inlets; check total inlet area.
  - ii) continuous inlets on both sides of barn are recommended.
- Fluctuating temperature and RH.
  - i) Remove 10-minute timers on fan controls.
  - ii) Use better quality thermostats with lower temperature differential.
  - iii) Use more smaller fans rather than fewer large fans.

CHAPTER 13 DESIGN EXAMPLES

DESIGN PROBLEM NO. 1

FREE STALL DAIRY BARN





Dimensions: 27,600 mm x 36,000mm
(92' x 120')
Insulated, wood frame construction
RSI 3.5 (R-20) in walls
RSI 4.9 (R-28) in ceilings

Stock: 110 mature dairy cattle

Location: Prince George

Required Information: Winter design temperature -33°C (-27°F)
Inside design temperature 0°C (32°F)
Building heat loss calculation:
17.6 kW (60,000 BTU/hr)
(floor, foundation, walls, ceiling)

Questions:
1. Recommended ventilation rates?
2. Is supplemental heat required?
3. What kind of ventilation system?

SOLUTION

1. Mature holstein cows at 640 kg (1400 lb); 640 kg cow over 450 kg cow in table 4.1 gives factor = 1.4.

   Winter ventilation rate 10 x 1.4 = 14 L/s (30 cfm) x 110 = **1540 L/s (3300 cfm)**

   Summer ventilation rate 140 x 1.4 = 196 L/s (415 cfm) x 110 = **21,560 L/s (45,700 cfm)**

2. Ventilation heat loss from equation 11.2 $q_v = (Q/v_a)c_p(t_i-t_o)$

   SI $$q_v = \frac{(1540\ L/s)(1000\ J/kg\text{-}°C)(0-(-33°C))}{(1000\ L/m^3)(0.78\ m^3/kg)} = 65,000\ J/s\ (or\ Watts)$$

   Imperial $$q_v = \frac{(3300\ ft^3/min)(0.24\ Btu/lb\text{-}°F)(32-(-27°F))(60\ min/hr)}{(12.5\ ft^3/lb)} = 224,000\ Btu/hr$$

   - Building heat loss is given q_b = 17,600 W (60,000 Btu/hr)
   - Sensible heat production from animals: table 3.1

     1.9 W/kg x 640 kg x 110 cows = 134,000 W (457,000 Btu/hr)
   - Therefore, supplemental heat required: $q_{sh} = q_v + q_b - q_s$

     q_{sh} = 65,000 + 17,600 - 134,000 W = **-51,400 W**

     = 224,000 + 60,000 - 457,000Btu/hr = **-173,000 Btu/hr**

   No supplemental heat is required. This heat balance indicates that, at this ventilation rate, the temperature will be higher than 0°C in the barn. Ventilation rate can be increased to maintain 0°C.

3. Recommended Ventilation System

   Option 1 - Poly-Tube System

   - Install two poly-tubes with driver fans, suspended above interior rows of stalls.
   - Specify 450 mm (18 in) diameter fans and tubes.
   - Specify 122 - 50 mm diameter holes (121 - 2 in Dia.) in two rows at the 3:00 and 9:00 o'clock positions. Hole spacing to be approximately 600 mm (24 in), on centres. From table 8.2.
   - Specify variable speed or two speed driver fans and equivalent exhaust fans; four fans in total.

- Specify separate controls for the two systems. Two-speed thermostat or modulator located in each of two sides of the barn.
- **Problems:** This system will ice-up in winter due to warmer interior air surrounding the cold poly-tube. The system will continue to operate but daily management will be required. Fogging may also occur.

Option 2

- Install conventional inlet flaps on exterior walls.
- Two 450 mm (18 in) exhaust fans; 2 speed or variable speed.
- **Problem:** Some fogging may occur.

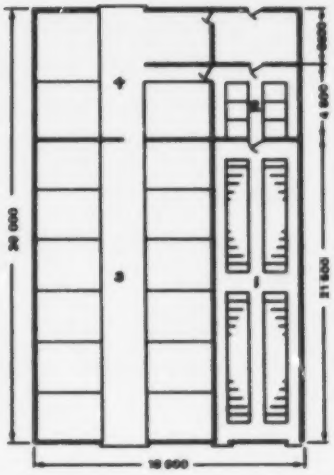
Option 3

- Consider installing 4 small heat exchangers sized for winter ventilation rates.
- **Problem:** Since no heating is required, it is not possible to justify expense on the basis of energy recovery.
- Advantages: Higher interior temperature. No fogging.

Note: Solutions do not account for summer ventilation. Open doors for natural ventilation. Pasture cows. Install larger poly-tube system to increase air exchange.

DESIGN PROBLEM NO. 2

CALF AND MATERNITY BARN



Dimensions: 18,000 mm x 30,000 mm (60' x 100')

(Area 1) 6,600 mm x 21,600 mm (22'x 72') - Nursery Room: 60 -65 kg (145 lb) (average) calves

60 - 600 mm x 1,500 mm (2' x 5') elevated calf stalls c/w slatted floors.

Insulated wood frame construction

RSI 3.5 (R-20) in walls

RSI 4.9 (R-28) in ceiling

Inside design temperature: 16°C (60°F)

Building heat loss calculation:

A/R = 132 W/°C (250 Btu/hr-°F)

(Area 2) 4,800 mm x 6,600 mm (16' x 22') - hospital room

6 - 1,200 mm x 1,800 mm (4' x 6') floor level box stalls

Insulated wood frame construction

RSI 3.5 (R-20) in walls

RSI 4.9 (R-28) in ceiling

Inside design temperature: 16°C (60°F)

Building heat loss calculation:

A/R = 37 W/°C (70 Btu/hr-°F)

Stock: 6 - 90 kg (200 lb) (average) Holstein calves

(Area 3) 11,400 mm x 21,600 mm (38' x 72') - maternity barn - 12 box stalls

(Area 4) 11,400 mm x 8,400 mm (38' x 28') - hospital area - 4 box stalls

wood frame construction

Insulation (optional) - cold environment

Location: Creston. Winter design temperature -21°C (-6°F)

Questions: 1. Recommended ventilation rates for calf rooms?

2. Is supplemental heat required in room 1?

3. What kind of ventilation/heating systems for all four areas?

SOLUTION

1. For room 1 with 60 - 65 kg (145 lb) calves:

60 x 5 L/s = **300 L/s (635 cfm)** - Winter

60 x 50 L/s = **3,000 L/s (6,350 cfm)** - Summer

For room 2 with 6 - 65 kg (145 lb) calves:

30 L/s **(64 cfm)** - Winter

300 L/s **(640 cfm)** - Summer

2. Ventilation heat loss from equation 11.2 $q_v = (Q/v_a)c_p(t_i - t_o)$

SI $q_v = \frac{(300\ L/s)(1{,}000\ J/kg°C)(16 - (-21°C))}{(1{,}000\ L/m^3)(0.83\ m^3/kg)} = 13{,}400\ W$

Imperial $q_v = \frac{(635\ ft^3/min)(0.24\ Btu/lb-°F)(60 - (-6°F)(60\ min/hr)}{(13.3\ ft^3/lb)} = 45{,}400\ Btu/hr$

Building heat loss A/R is given.

SI $q_b = (132\ W/°C(16 - (-21°C)) = 4{,}900\ W$

Imperial $q_b = (250\ Btu/hr-°F)(60 - (-6°F)) = 16{,}500\ Btu/hr$

Sensible heat production from table 3.1

1.4 W/kg x 65 kg x 60 calves = 5,400 W (18,600 Btu/hr)

Therefore, supplemental heat required: $q_{sh} = q_v + q_b - q_s$

q_{sh} = 13,400 + 4,900 - 5,400 W = **12,900 W (44,000 Btu/hr)**

Specify combination of unit heaters. Heat lamps or radiant heaters up to 12,900 Watts.

3. Area 1

- Specify poly-tube inlet duct, or equivalent, suspended from ceiling over centre alley. Air to discharge downwards if calves are tethered head-to-head.
- Specify 300 mm (12 in) diameter duct. (Possible to go to 450 mm (18 in) dia. duct with variable speed fan.)
- Specify 216 - 25 mm (1 in) diameter holes in two rows at 5:30 and 6:30 o'clock positions. Hole spacing to be approximately 200 mm (8 in), on centres.
- Specify variable speed or 2-speed driver fan and exhaust fan.
- Fresh air duct could be constructed in attic space. Air inlet could be as simple as a 12 mm (½ in) slot in ceiling, discharging fresh air straight down into walkway.
- For spring/summer ventilation, install conventional air inlet on exterior wall. Specify exhaust fans and controls to 3,000 L/s (6,350 cfm). Say, 2 - 450 mm (18") Danor S-18E fans, or equal.

Area 2

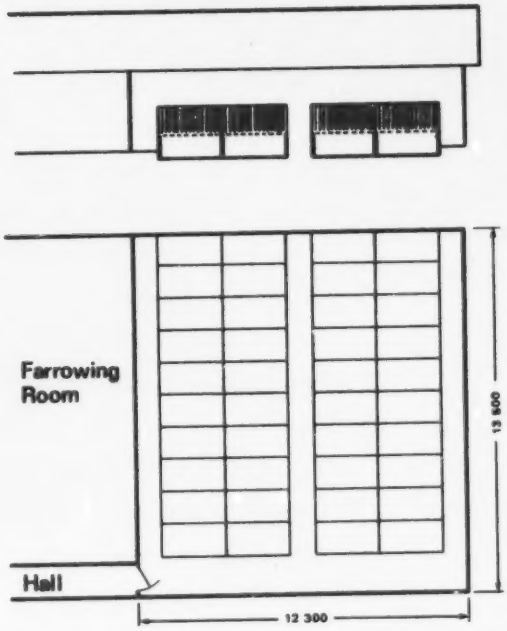
- Specify rigid air inlet duct discharging air down into centre walkway. Minimum duct cross-section area - .06 m^2 (.64 ft^2).
- Specify Danor S8-2 fan, or equal, for winter ventilation and Danor S-10, or equal, for summer.

Areas 3 and 4

- Specify natural ventilation. Continuous ridge vent opening to be 200 mm (8 in) wide. Specify "Tensar" windbreak cladding on outside wall.

DESIGN PROBLEM NO. 3

WEANLING ROOM



Dimensions: 12,300 mm x 13,500 mm
(41' x 45')
Insulated, wood frame construction
RSI 4.9 (R-28) in ceiling and
RSI 3.5 (R-20) in walls
40 - 1,200 mm x 2,400 mm (4' x 8') decks
Inside design temperature: 25°C (77°F)
Building heat loss calculation:
A/R = 103 W/°C (195 Btu/hr-°F)
Stock: 500 weaner pigs average weight = 10 kg (22 lb)

Location: Abbotsford. Winter design temperature: -10°C (14°F)
(from Appendix - Design Data)

Questions:
1. Recommended ventilation rates?
2. How much supplemental heat?
3. What kind of ventilation/heating system?

SOLUTION

1. Winter ventilation rate: 500 pigs x 0.9 L/s/pig = **450 L/s (950 cfm)**
   Summer ventilation rate: 500 pigs x 15 L/s/pig = **7,500 L/s (15,900 cfm)**

2. Supplemental heat $q_{sh} = q_v + q_b - q_s$; $q_v = (Q/vs)cp(t_i - t_o)$ (Eq. 11.2)

$$q_v = \frac{(450\ L/s)(1{,}000\ J/kg°C)(25 - (-10°C))}{(1{,}000\ L/m^3)(0.87\ m^3/kg)} = 18{,}100\ W\ (61{,}800\ Btu/hr)$$

q_b = (103 W/°C)(25 - (-10°C)) = 3,605 W (12,300 Btu/hr)

q_s = (3.1 W/kg)(10 kg/pig)(500 pigs) = 15,500 W (52,900 Btu/hr)

q_{sh} = 18,100 + 3,605 - 15,500 W = **6,205 W (21,200 Btu/hr)**

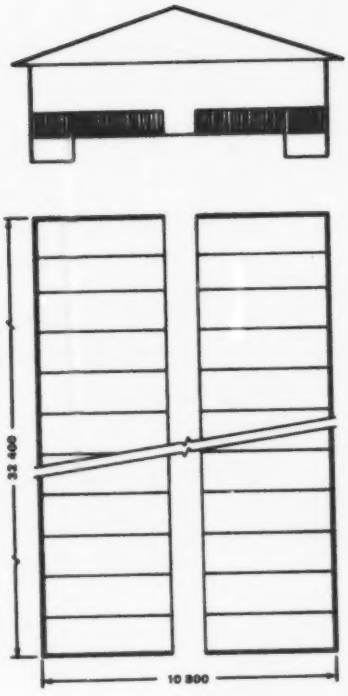
Specify heat lamps, electric unit heaters, electric radiant heaters or draw fresh air from pre-heated hallway.

3. Recommended Ventilation System

- Note that individual, all-in-all-out weaner rooms are preferable to a one-room continuous nursery operation as in this illustration.
- Specify two adjustable, side air-inlets for air mixing and distribution. Inlets to be located on 13,500 mm walls. Fresh air to be drawn from insulated duct on interior wall.
- Specify a two speed fan (450 L/s and 900 L/s) for winter ventilation c/w S.P.D.T. thermostat control.
- Specify three, 2,500 L/s single speed fans c/w three thermostats with set points of 26°C, 27°C and 28°C.

DESIGN PROBLEM NO. 4

FINISHING BARN



Dimensions: 10,800 mm x 32,400 mm (36' x 108')
Insulated, wood frame construction
RSI 4.9 (R-28) in ceiling and
RSI 3.5 (R-20) in walls
38 mm (1½") poly-styrene insulation
on foundation walls
28 - 2,290 x 4,800 (7'6" x 16'0")
pens: 14 - 80 kg hogs per pen
(14 x 28 = 392 hogs total)
Inside design temperature: 16°C (60°F)
Building heat loss calculation:
A/R = 190 W/°C (360 Btu/hr-°F)

Location: Dawson Creek. Winter design temperature:
-36°C (-33°F)

Questions: Recommended ventilation rates?
Is supplemental heat required?
What kind of ventilation system?

SOLUTION

1. Winter ventilation rate: 392 x 3.3 = **1,290 L/s (2,740 cfm)**
   Summer ventilation rate: 392 x 38 = **14,900 L/s (31,600 cfm)**

2. Supplemental heat $q_{sh} = q_v + q_b - q_s$; $q_v = (Q/v_s)c_p(t_i - t_o)$ (Eq. 11.2)

$$q_v = \frac{(1{,}290\ L/s)(1{,}000\ J/kg\text{-}°C)(16 - (-36°C))}{(1{,}000\ L/m^3)(0.83\ m^3/kg)} = 80{,}800\ W\ (276{,}000\ Btu/hr)$$

$$q_b = (190\ W/°C)(16 - (-36°C)) = 9{,}900\ W\ (33{,}700\ Btu/hr)$$

$$q_s = (1.5\ W/kg)(80\ kg/pig)(392\ pigs) - 47{,}000\ W\ (160{,}000\ Btu/hr)$$

q_{sh} = 80,800 + 9,900 - 47,000 W = **43,700 W (149,000 Btu/hr)**

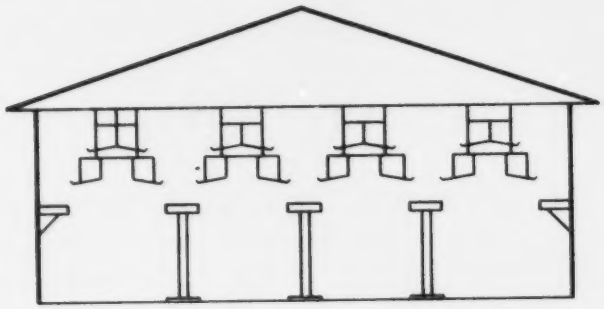
At this ventilation rate and inside temperature there is a heat deficit. Options include reducing ventilation rate by half, reducing inside temperature or installing supplemental heaters. Heat exchangers may prove to be effective.

3. Recommended Ventilation System

- Specify centre, adjustable air inlet system.
- Specify two, two-speed 650 L/s / 1,300 L/s fans c/w S.P.D.T. thermostats set at 10°C and 16°C.
- Specify two single speed 2,500 L/s fans c/w S.P.S.T. thermostats set at 17°C.
- Select two single speed 4,500 L/s fans c/w S.P.S.T. thermostats set at 18°C.

DESIGN PROBLEM NO. 5

DEEP PIT LAYER BARN



Dimensions: 13,200 mm x 60,000 mm
(44' x 200')
Insulated, wood frame construction
RSI 4.9 (R-28) in ceiling,
RSI 3.5 (R-20) in walls
38 mm (1½") polystyrene insulation on foundation walls
Inside design temperature:
18°C (65°F)
Building heat loss calculation
A/R = 430 W/°C (820 Btu/hr-°F)
Stock: 4 - 54,000 mm (180') cage-rows
3 birds per 300 mm x 450 mm (12" x 18") cage = 8,640 birds

Location: Cranbrook. Winter design temperature: -17°C (-27°F)

Questions:
1. Recommended ventilation rates?
2. Is supplemental heating required?
3. What kind of ventilation/heating system?

SOLUTION

1. Winter ventilation rate: 8,640 x 0.25 = **2,160 L/s (4,580 cfm)**
   Summer ventilation rate: 8,640 x 3.5 = **30,240 L/s (64,000 cfm)**

2. Supplemental heat: $q_{sh} = q_v + q_b - q_s$; $q_v = (Q/v_s)cp(t_i - t_o)$ (Eq. 11.2)

$$q_v = \frac{(2,160\ L/s)(1,000\ J/kg\text{-}°C)(18 - (-27°C))}{(1,000\ L/m^3)(0.84 m^3/kg)} = 115,700\ W\ (395,000\ Btu/hr)$$

q_b = 430 W/°C (18 - (-27°C)) = 19,300 W (66,000 Btu/hr)

q_s = (3.9 W/kg)(1.8 kg/bird)(8,640 birds) = 60,600 W (207,000 Btu/hr)

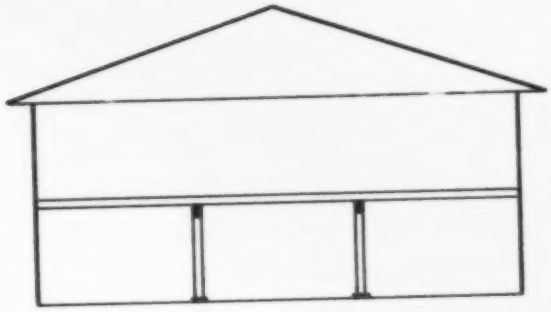
q_{sh} = 115,700 + 19,300 - 60,000 W = **74,400 W (254,000 Btu/hr)**

Supplemental heating is required at this ventilation rate and temperatue difference. Specify hot water heating c/w black steel pipe radiators around perimeter of barn.

3. Recommended ventilation system
   - Specify two, side air-inlets on both sides of barn
   - Specify two, 1,100 L/s single speed fans for winter ventilation
   - Specify five 5,600 L/s single speed fans on five thermostats stepped sequentially
   - Specify light control fan housings

DESIGN PROBLEM NO. 6

BROILER BARN



Dimensions: 12,000 mm x 60,000 mm
(40' x 200')
Insulated, wood frame construction
RSI 4.9 (R-28) in ceiling and RSI 3.5 (R-20) in walls
Inside design temperature (brooding temperature):
29°C (85°F)
Building heat loss factor:
A/R = 458 W/°C (870 Btu/hr-°F)
Stock: 2 x 10,600 birds
@ .070 m^2 (.75 ft^2)/bird

Location: Salmon Arm. Winter design temperature: -23°C (-9°F)

Questions:
1. Recommended ventilation rates?
2. How much supplemental heat?
3. What kind of heating/ventilation systems?

SOLUTION

1. Winter ventilation rates: 10,600 x .05 = 530 L/s (1,120 cfm) and up to 2,650 L/s (5,610 cfm) for mature birds
   Summer ventilation rate: 10,600 x 25,440 L/s (54,000 cfm)

2. Supplemental heat $q_{sh} = q_v + q_b - q_s$; $q_v = (Q/v_s)c_p(t_i - t_o)$ (Eq. 11.2)

$$q_v = \frac{(530\ L/s)(1{,}000\ J/kg\text{-}°C)(29 - (-23°C))}{(1{,}000\ L/m^3)(0.88\ m^3/kg)} = 31{,}300\ W\ (107{,}000\ Btu/hr)$$

q_b = (458)(29 - (-23)) = 23,800 W (81,200 Btu/hr)

q_s = (12 W/kg) (0.04 kg)(10,600) = 5,090 W (17,300 Btu/hr)

q_{sh} = 31,300 + 23,800 - 5,090 = **50,000 W (171,000 Btu/hr)**

For brooding chicks, 50,000 W supplemental heat is required. Specify eight, 7000 W brooders or equivalent.

Note that the heat balance for mature birds only requires 38,000 W at the recommended ventilation rate and at an interior temperature of 22°C.

3. Recommended ventilation system

- Specify two, side air-inlets on both sides of barn; both floors of barn
- Specify two, 1,400 L/s variable speed fans for winter ventilation on one modulator with minimum speed set to 20% of maximum for brooding period. Adjust minimum speed setting upwards as birds grow.
- Specify four, 3,000 L/s variable speed fans on two modulators in series with two thermostats. Adjust set-points downwards as birds grow.
- Specify two, 6,000 L/s single speed fans on two thermostats. Adjust set-points downwards as birds grow.

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306.400-2	Environmental Control for Reducing Energy Waste in Farm Buildings (OH)
306.406-1	Troubleshooting the Ventilating System (SA)
306.407-1	Manure Pit Ventilation in Confinement Livestock Buildings (PU)
306.435-1	Maintenance & Operation of Ventilating Fans for Hog Barns (MSU)
306.435-2	The Environment in Swine Housing (MSU)
306.450-1	Livestock Building Exhaust Fans (SA)
306.500-1	Insulation in Farm Buildings (AC)
310.000-5	Beef Handbook - Housing and Equipment (MWPS)
320.100-4	Dairy Handbook - Housing and Equipment (MWPS)
350.000-1	Sheep Handbook - Housing and Equipment (MWPS)
360.000-1	Swine Handbook - Housing and Equipment (MWPS)
362.000-1	Confinement Swine Housing (AC)
362.001-1	Floor Heat for Swine (MSU)
362.001-2	Spray Cooling of Feeder Pigs (OMAF)
373.500-1	Horse Handbook - Housing and Equipment (MWPS)
380.750-1	Manure Gases (SA)
381.610-1	Developing Good Dunging Habits of Swine on Partially Slotted Floors (BCMAF)
386.100-1	Environmental Guidelines for Dairy Producers (BCMAF)
386.100-2	Environmental Guidelines for Poultry Producers (BCMAF)
386.100-3	Environmental Guidelines for Swine Producers (BCMAF)
386.100-4	Environmental Guidelines for Beef Producers (BCMAF)

Canada Plan Service (CPS) Auxiliary Plans Available from B.C. Ministry of Agriculture and Food

306.41-1	Natural Ventilation - Ridge Vent & Hinged Wall Panels
M9341 (L)	Sliding Doors
M9351 (L)	Plastic Curtain Wall Ventilation
9701 (L)	Interlocked Heating/Ventilating Control for LIvestock Buildings
Q9713	Centre Ceiling Air Inlet
Q9714	Adjustable Side Air Inlet

KEY:

AC	Agriculture Canada
BCMAF	B.C. Ministry of Agriculture and Food
MSU	Michigan State University
MWPS	Midwest Plan Service
OH	Ontario Hydro
PU	Purdue University
SA	Saskatchewan Agriculture

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APPENDIX

DESIGN DATA FOR SELECTED LOCATIONS IN CANADA

Province and Location	Design Temperature				Degree Days Below 18°C	15 Min. Rain., mm	One Day Rain., mm	Ann. Tot. Prcp., mm	Gnd. Snow Load, kN/m²	Hourly Wind Pressures			Seismic Data	
	January		July 2½%											
	2½%, °C	1%, °C	Dry, °C	Wet, °C						1/10, kN/m²	1/30, kN/m²	1/100, kN/m²	Zone	Acceleration Ratio, A
British Columbia														
Abbotsford	-10	-11	29	20	3 148	10	83	1 502	2.4	0.42	0.55	0.71	3	0.08
Agassiz	-13	-15	31	20	2 950	8	116	1 648	3.1	0.55	0.75	1.00	2	0.04
Alberni	-5	-7	31	18	3 160	10	125	2 019	2.6	0.47	0.58	0.70	3	0.08
Ashcroft	-25	-28	34	20	3 687	10	45	224	1.3	0.28	0.35	0.43	1	0.02
Beaton River	-37	-39	25	18	7 068	13	50	435	3.2	0.22	0.27	0.34	1	0.02
Burns Lake	-30	-33	25	17	5 801	10	48	513	2.5	0.30	0.36	0.43	2	0.04
Cache Creek	-25	-28	34	20	3 800	10	63	250	1.4	0.29	0.35	0.43	1	0.02
Campbell River	-7	-9	26	18	3 417	10	105	1 539	2.8	0.46	0.58	0.72	3	0.08
Carmi	-24	-26	33	20	5 191	10	98	563	3.4	0.24	0.33	0.44	1	0.02
Castlegar	-19	-22	32	20	3 751	10	51	710	3.4	0.23	0.30	0.39	0	0
Chetwynd	-35	-38	27	18	6 100	15	63	410	2.2	0.32	0.37	0.44	1	0.02
Chilliwack	-12	-13	30	20	2 932	8	122	1 741	2.8	0.48	0.63	0.83	2	0.04
Cloverdale	-8	-10	29	20	3 080	8	102	1 270	2.1	0.46	0.58	0.72	3	0.08
Comox	-7	-9	27	18	3 203	10	113	1 206	2.5	0.45	0.58	0.74	3	0.08
Courtenay	-7	-9	28	18	3 205	10	105	1 452	2.5	0.45	0.58	0.74	3	0.08
Cranbrook	-27	-30	32	19	4 762	10	43	438	2.4	0.22	0.29	0.37	0	0
Crescent Valley	-20	-23	31	19	4 228	10	52	786	3.4	0.22	0.29	0.37	0	0
Crofton	-6	-8	28	18	2 850	8	76	1 029	2.0	0.48	0.58	0.69	3	0.08
Dawson Creek	-36	-39	27	18	6 152	18	67	425	2.0	0.31	0.37	0.44	1	0.02
Dog Creek	-28	-30	29	18	5 207	10	47	393	1.9	0.31	0.37	0.44	1	0.02
Duncan	-6	-8	29	18	2 907	8	110	1 043	2.0	0.48	0.58	0.69	3	0.08
Elko	-28	-31	29	19	4 379	13	54	580	3.5	0.27	0.37	0.50	0	0
Fernie	-29	-32	29	19	4 970	13	106	1 082	4.6	0.33	0.43	0.55	0	0
Fort Nelson	-40	-42	28	18	7 064	13	81	446	2.4	0.19	0.24	0.29	1	0.02
Fort St. John	-36	-38	26	18	6 118	15	80	450	2.5	0.31	0.36	0.42	1	0.02
Glacier	-27	-30	27	17	5 799	10	71	1 493	7.6	0.24	0.29	0.35	1	0.02
Golden	-28	-31	29	17	4 869	8	59	473	3.8	0.27	0.32	0.38	0	0
Grand Forks	-20	-22	35	20	4 037	10	41	450	2.0	0.26	0.36	0.48	1	0.02
Greenwood	-20	-22	35	20	4 506	10	107	478	1.9	0.29	0.39	0.52	1	0.02
Hazey	-9	-11	30	20	2 946	10	117	1 926	2.3	0.47	0.60	0.77	3	0.08
Hope	-16	-18	32	20	3 128	8	106	1 601	3.4	0.41	0.55	0.73	2	0.04
Kamloops	-25	-28	34	20	3 735	13	57	268	1.8	0.30	0.37	0.45	1	0.02
Kaslo	-23	-26	29	19	4 070	10	51	785	3.0	0.22	0.28	0.36	0	0
Kelowna	-17	-20	33	20	3 647	10	64	320	1.9	0.34	0.43	0.53	1	0.02
Kimberley	-26	-29	31	19	4 840	10	49	378	3.0	0.22	0.29	0.37	0	0
Kitimat Plant	-16	-18	23	16	4 103	13	185	2 826	3.5	0.22	0.26	0.31	3	0.08
Kitimat Townsite	-16	-18	23	16	4 187	13	119	2 377	4.5	0.22	0.26	0.31	3	0.08
Langley	-8	-10	29	20	3 127	8	118	1 501	2.2	0.45	0.58	0.73	3	0.08
Lillooet	-23	-25	33	20	4 000	10	114	391	2.5	0.32	0.39	0.49	1	0.02
Lytton	-19	-22	35	20	3 239	10	77	463	3.0	0.31	0.39	0.49	1	0.02
Mackenzie	-35	-38	26	17	6 150	10	63	430	3.6	0.24	0.29	0.35	1	0.02
McBride	-34	-37	30	18	4 938	13	50	525	3.4	0.27	0.32	0.38	1	0.02
McLeod Lake	-35	-37	27	17	5 867	10	63	460	2.5	0.24	0.29	0.35	1	0.02
Merritt	-7	-9	17	15	3 726	13	76	1 409	1.8	0.49	0.58	0.68	3	0.08
Merritt	-26	-29	34	20	4 325	8	57	254	2.0	0.32	0.39	0.49	1	0.02
Mission City	-9	-11	30	20	2 956	13	98	1 573	2.4	0.47	0.60	0.77	3	0.08
Montrose	-17	-20	32	20	4 088	10	51	630	3.2	0.22	0.30	0.41	0	0
Nakusp	-24	-27	31	19	4 037	10	51	790	3.6	0.24	0.30	0.37	0	0
Nanaimo	-7	-9	26	18	3 304	8	92	1 085	2.6	0.47	0.58	0.71	3	0.08
Nelson	-20	-24	31	19	3 669	10	66	763	3.3	0.22	0.29	0.37	0	0
New Westminster	-8	-10	29	19	2 948	10	132	1 520	2.1	0.44	0.55	0.68	3	0.08
North Vancouver	-7	-9	26	19	3 080	10	100	1 791	2.2	0.44	0.55	0.68	3	0.08
Ocean Falls	-12	-14	23	16	3 499	13	234	4 390	3.0	0.47	0.55	0.65	3	0.08
100 Mile House	-28	-31	30	18	4 500	10	51	460	2.6	0.30	0.36	0.43	1	0.02
Osoyoos	-16	-18	33	20	3 295	10	35	342	1.4	0.30	0.43	0.59	1	0.02
Penticton	-16	-18	33	20	3 513	10	45	296	1.3	0.40	0.52	0.68	1	0.02
Port Alberni	-5	-7	31	18	3 312	10	140	2 009	2.6	0.47	0.58	0.70	3	0.08
Port Hardy	-5	-7	20	16	3 659	13	131	1 730	2.1	0.49	0.58	0.68	3	0.08
Port McNeill	-5	-7	22	17	3 493	13	127	1 270	2.4	0.49	0.58	0.68	3	0.08
Powell River	-9	-11	26	18	3 307	8	88	1 017	2.4	0.42	0.55	0.71	3	0.08
Column 1	2	3	4	5	6	7	8	9	10	11	12	13	14	15

From: National Building Code. 1980. Associate Committee on the National Building Code. Ottawa, Ontario: National Research Council.

DESIGN DATA FOR SELECTED LOCATIONS IN CANADA

Province and Location	Design Temperature				Degree Days Below 18°C	15 Min. Rain., mm	One Day Rain., mm	Ann. Tot. Pcpn., mm	Gnd. Snow Load, kN/m²	Hourly Wind Pressures			Seismic Data	
	January		July 2½ %											
	2½ %, °C	1%, °C	Dry, °C	Wet, °C						1/10, kN/m²	1/30, kN/m²	1/100, kN/m²	Zone	Acceler-ation Ratio, A
Prince George	-33	-36	28	18	5 387	15	50	621	2.6	0.25	0.30	0.36	1	0.02
Prince Rupert	-14	-16	19	15	4 383	13	141	2 415	2.6	0.42	0.50	0.59	3	0.08
Princeton	-27	-30	32	20	4 554	10	37	359	2.3	0.24	0.32	0.42	1	0.02
Qualicum Beach	-7	-9	27	18	3 205	10	102	1 270	2.6	0.46	0.58	0.72	3	0.08
Quesnel	-33	-35	30	17	4 733	10	72	514	2.7	0.25	0.29	0.34	1	0.02
Revelstoke	-26	-29	32	19	4 073	13	78	1 096	4.6	0.24	0.29	0.35	2	0.04
Richmond	-7	-9	27	19	2 920	8	114	1 450	1.9	0.45	0.55	0.67	3	0.08
Salmon Arm	-23	-26	33	20	3 894	13	43	531	2.8	0.29	0.35	0.43	1	0.02
Sandspit	-6	-7	15	15	3 681	13	80	1 261	2.3	0.54	0.63	0.74	3	0.08
Sidney	-6	-8	26	18	2 800	8	102	790	1.6	0.46	0.55	0.66	3	0.08
Smithers	-29	-31	25	17	5 305	13	60	502	2.2	0.31	0.37	0.44	2	0.04
Smith River	-46	-48	26	17	7 652	8	68	465	2.8	0.19	0.25	0.33	2	0.04
Squamish	-11	-13	29	20	3 314	10	112	2 061	3.2	0.38	0.50	0.65	3	0.08
Stewart	-23	-25	23	16	4 701	13	178	1 843	8.4	0.32	0.39	0.48	3	0.08
Taylor	-36	-38	26	18	6 100	15	56	398	2.5	0.32	0.37	0.44	1	0.02
Terrace	-20	-22	25	16	4 429	13	117	1 200	5.1	0.36	0.43	0.51	2	0.04
Tofino	-2	-4	19	16	3 217	13	174	3 061	2.5	0.54	0.63	0.74	3	0.08
Trail	-17	-20	33	20	3 639	10	51	664	3.2	0.17	0.24	0.33	0	0
Ucluelet	-2	-4	19	16	3 180	13	140	2 690	2.4	0.54	0.63	0.74	3	0.08
Vancouver	-7	-9	26	19	3 005	10	94	1 324	1.9	0.45	0.55	0.67	3	0.08
Vernon	-20	-23	33	20	3 838	13	40	393	2.0	0.32	0.39	0.49	1	0.02
Victoria	-5	-7	24	17	3 075	5	81	657	1.5	0.48	0.58	0.70	3	0.08
Williams Lake	-31	-34	29	17	4 625	10	37	402	2.9	0.30	0.35	0.41	1	0.02
Youbou	-5	-7	31	19	2 942	10	114	1 650	2.5	0.46	0.55	0.66	3	0.08
Alberta														
Athabasca	-35	-38	28	19	6 310	18	88	442	2.2	0.30	0.37	0.45	0	0
Banff	-30	-32	27	17	5 719	18	53	477	2.8	0.39	0.45	0.52	0	0
Barrhead	-34	-37	28	19	6 000	20	102	460	2.2	0.32	0.39	0.49	0	0
Beaverlodge	-35	-38	28	18	5 909	25	101	455	2.1	0.27	0.33	0.40	0	0
Brooks	-32	-34	32	19	5 266	18	89	353	1.6	0.39	0.48	0.57	0	0
Calgary	-31	-33	29	17	5 344	23	95	437	0.9	0.40	0.46	0.54	0	0
Campsie	-34	-37	28	19	6 080	20	111	445	2.3	0.32	0.39	0.49	0	0
Camrose	-33	-35	29	19	5 960	20	92	394	1.7	0.21	0.29	0.39	0	0
Cardston	-30	-33	29	18	4 870	20	102	495	1.8	0.74	0.93	1.15	0	0
Claresholm	-31	-34	29	18	4 782	15	97	460	1.0	0.66	0.80	0.96	0	0
Cold Lake	-36	-38	28	20	6 183	15	94	433	1.5	0.31	0.37	0.44	0	0
Coleman	-31	-34	28	18	5 418	15	62	537	2.6	0.54	0.69	0.87	0	0
Coronation	-31	-33	30	19	5 905	20	99	373	1.9	0.23	0.32	0.43	0	0
Cowley	-31	-34	29	18	5 189	15	74	504	1.7	0.73	0.91	1.13	0	0
Drumheller	-31	-33	29	18	5 330	20	73	358	1.6	0.32	0.39	0.49	0	0
Edmonton	-32	-34	28	19	5 590	23	114	446	1.5	0.32	0.40	0.51	0	0
Edson	-34	-37	28	18	5 962	18	79	554	2.4	0.36	0.43	0.50	0	0
Embarras Portage	-41	-44	27	19	6 999	10	82	399	1.6	0.31	0.37	0.45	0	0
Fairview	-38	-40	27	18	6 152	15	64	422	2.1	0.26	0.32	0.39	0	0
Fort McMurray	-39	-41	28	19	6 779	13	61	435	1.8	0.27	0.32	0.38	0	0
Fort Saskatchewan	-32	-35	28	19	5 881	20	78	430	1.5	0.31	0.39	0.49	0	0
Fort Vermilion	-41	-43	28	18	7 077	13	60	360	2.0	0.22	0.26	0.32	0	0
Grande Prairie	-36	-39	27	18	6 144	23	78	442	2.1	0.37	0.44	0.52	0	0
Habay	-41	-43	28	18	7 050	13	63	360	2.5	0.20	0.24	0.28	0	0
Hardisty	-33	-35	30	19	5 950	20	56	384	1.8	0.24	0.32	0.42	0	0
High River	-31	-33	28	17	5 414	18	111	489	1.9	0.51	0.60	0.72	0	0
Jasper	-32	-35	28	18	5 530	10	108	402	2.4	0.37	0.43	0.50	0	0
Keg River	-40	-42	28	18	6 854	13	60	403	2.6	0.19	0.24	0.29	0	0
Lac La Biche	-35	-38	28	19	6 308	15	82	460	2.1	0.31	0.37	0.44	0	0
Lacombe	-33	-35	29	18	5 761	23	71	456	1.8	0.24	0.31	0.40	0	0
Lethbridge	-30	-33	31	18	4 717	20	93	436	1.5	0.64	0.76	0.91	0	0
Manning	-39	-41	27	18	6 600	13	51	360	2.5	0.21	0.26	0.32	0	0
Medicine Hat	-31	-34	33	19	4 872	23	122	348	1.4	0.39	0.49	0.60	0	0
Peace River	-37	-40	27	18	6 424	15	48	351	2.4	0.24	0.29	0.36	0	0
Penhold	-32	-35	29	18	5 845	23	124	449	1.5	0.31	0.37	0.44	0	0
Column 1	2	3	4	5	6	7	8	9	10	11	12	13	14	15